COMBUSTION

EVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

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VOLUME SIX NUMBER THREE

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FOR SEPTEMBER 1934

FEATURE ARTICLES

EDITORIALS

DEPARTMENTS

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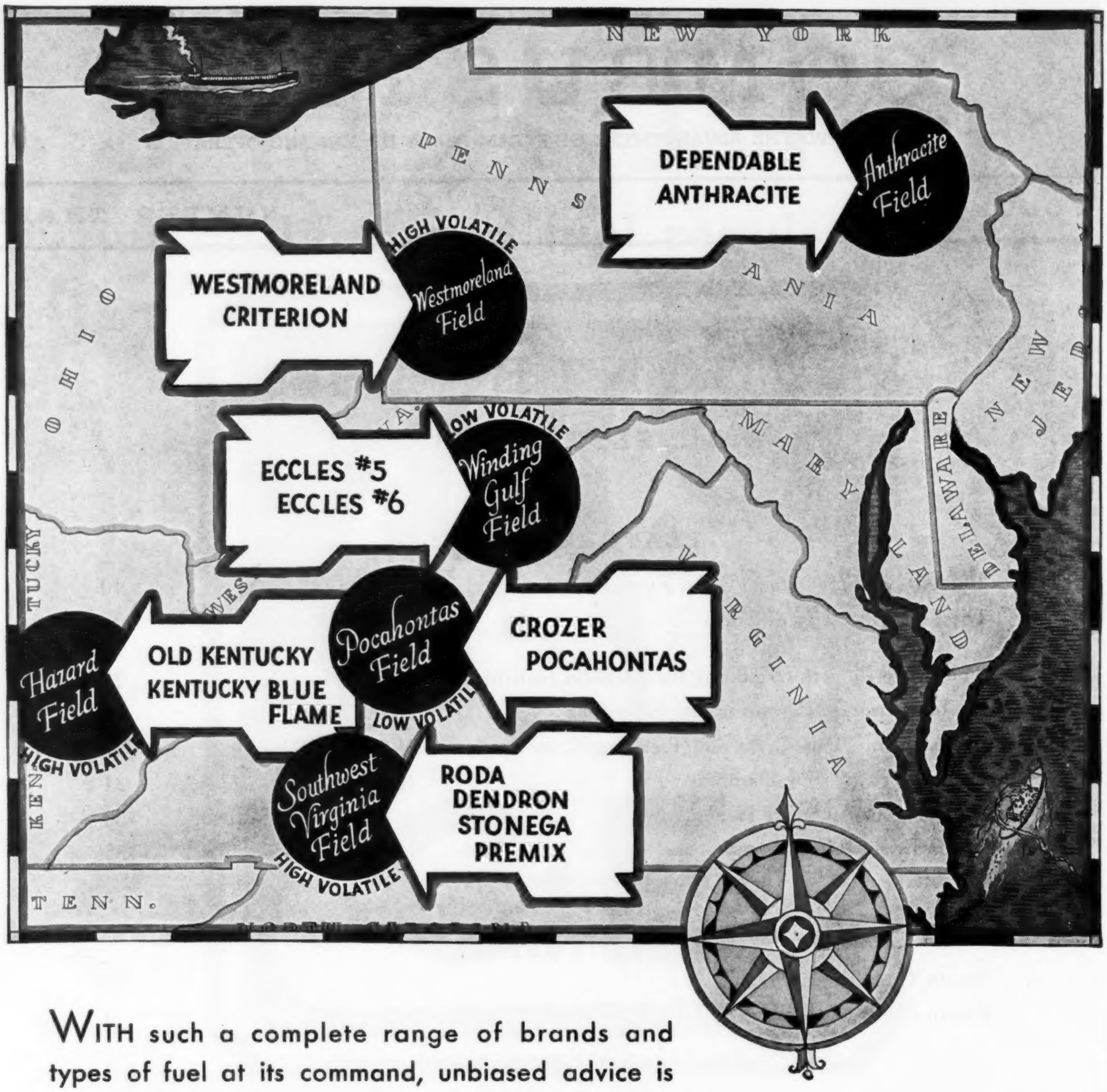

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EDITORIAL

Steam Table Research

The Third International Steam Table Conference will be held in this country September 17 to 22, inclusive, under the auspices of the American Society of Mechanical Engineers with sessions at the Bureau of Standards, Washington, the Massachusetts Institute of Technology, Cambridge, Mass., and the Engineering Societies Building, New York.

Credit for the comprehensive research into the properties of steam that have been going on for the past dozen years belongs to a small group of scientists and engineers headed by George Orrok, who, conscious of trends in design for higher pressures and temperatures and noting the wide descrepancies among existing steam tables in the higher pressure and superheat range, initiated the program in 1921, under the auspices of the A.S.M.E. Special Research Committee on the Thermal Properties of Steam. These investigations, it will be recalled, were allocated to the Bureau of Standards, under the direction of Dr. H. S. Osborne; the Massachusetts Institute of Technology under Dr. F. G. Keys and the Harvard Engineering School under Dr. Harvey N. Davis and R. V. Kleinschmidt. The first named was assigned the task of determining the latent heats of evaporation and heat of water; the second was given the problem of determining saturation temperatures and volumes atvarious pressures and superheat, and the last the superheat values.

Simultaneously with this work investigations were being carried on abroad by Callendar in England, Hausen in Germany and others. The first steam tables based on the American investigations were compiled by Professor Keenan and published by the A.S.-M.E., first in tentative form and later in more comprehensive form in 1930.

It was found that lack of agreement existed in the values obtained by the several investigators and the First International Steam Table Conference was called in England in the summer of 1929 to discuss and, if possible, adjust these discrepancies. The outcome of this conference was agreement on a skeleton table and the establishment of certain tolerances subject to subsequent revision; also agreement as to the definition of the international kilowatt hour as equivalent to 860 kilocalories.

At the close of the World Power Conference in Berlin in 1930 a Second Steam Table Conference was held and further tolerances of about 1 part in 700 were established. It is hoped that, as a result of subsequent research both here and abroad, that the present conference will be able further to reduce these tolerances to about 1 part in 3000 or 4000. The values most affected will be in the superheat and critical regions. It is contemplated that revision of the steam tables will be started this fall and completed in about three years.

While the older steam tables are sufficiently accurate for the majority of ordinary power plant computations, design is constantly pushing forward into new regions and it is highly desirable that all doubt be removed as to values in the higher pressure and superheat regions so that extrapolated values as well as actual values may be in substantial agreement.

It has been said that engineering knows no national borders, yet economic conditions in different countries exert a strong influence on engineering practice. On the other hand, the properties of steam are universal, and, granting equal accuracy of observations, discrepancies in the results of different investigators must be attributable to methods. That the values are likely to be brought into line for all practical purposes represents a monumental accomplishment well worth the money and effort put into the work.

Cheap Power?

To date the PWA power grants total more than 267 million dollars, and the end is nowhere in sight. The bulk of this money, exclusive of TVA, is being expended for hydro developments in regions that cannot hope to provide adequate markets for the large blocks of power that will be made available.

According to the President, the objective is *cheap power*. Cheap power, perhaps, for the few in the favored localities at the expense of the taxpayers as a whole. The vast expenditures must be met indirectly, in some form or another, by all power consumers and may be considered as equivalent to adding to their existing power bills. That the money will not swell the receipts of the utilities may be solace to some individuals but the effect on the public pocketbook is the same.

All this we are told is for the purpose of establishing "yardsticks"—a term calculated to appeal to the popular imagination, but meaningless to any one who knows something of the economics of power production, with the innumerable factors that affect the cost.

It is true that some far-sighted utility managements have come to the conclusion that lower rates are desirable as an effective means of building load through wider use of electrical appliances. This, however, is a matter that must be governed by local conditions rather than any nation-wide program.

Opposition to the Government's program has heretofore been regarded as biased, but it is high time that engineering opinion become crystallized and be exerted to offset the views of certain well meaning but poorly informed advisers on Governmental power policies.

Pulverized Anthracite as Boiler Fuel

at the New Tir John Station, Swansea, England

Studies leading up to the adoption of pulverized anthracite for the latest large British power station are discussed by our London correspondent and a description of the steam generating equipment included. Lack of markets for anthracite duff from the South Wales fields has rendered this a very cheap fuel for power generation and the experience at Tir John Station following its completion in November should greatly influence its wider use in pulverized form.

THE chief source of anthracite in Great Britain is in the South Wales coal fields from which the bulk of coal produced is of a high grade steam quality with low volatile content. Formerly the principal outlet for South Wales coals was the marine trade and, consequently, this area has been severely hit by the slump in shipping activities coupled with the competition of fuel oils. As a result, the coal operators have been faced with the problem of finding new markets and have shown particular interest in pulverized fuel firing both with a view to securing a share of the market offered by the many capital power stations utilizing pulverized fuel and also with the object of reducing production costs.

Two of the principal groups, the Amalgamated Anthracite Collieries Ltd. and Powell Duffryn Steam Coal Co. Ltd. converted old boiler plants or installed new steam raising equipment, and are utilizing colliery refuse in pulverized form for their own power and steam requirements. In both cases the fuel employed is anthracite duff, which has been accumulating over many years, augmented by the rejects from dry cleaning plants, etc., for which there has previously been practically no outlet except as a hand-fired fuel in small quantities.

As a result of the experience obtained from these more or less experimental installations the new super-power station at Tir John, Swansea, which will form the principal station in the South Western area under the Central Electricity Board's "Grid" scheme, has been designed to burn anthracite duff. The Corporation has entered into a contract with one of the principal local colliery groups for the purchase of this class of fuel over a period of twenty years at a price which represents little more than the bare cost of handling and transport. The consulting engineers, Messrs. Preece, Cardew & Rider, investigated the possibilities very closely before making their recommendations and designing the new plant, which, as the latest station to be planned and constructed in Great Britain, may be regarded as an epitome of pulverized fuel firing experience in that country.

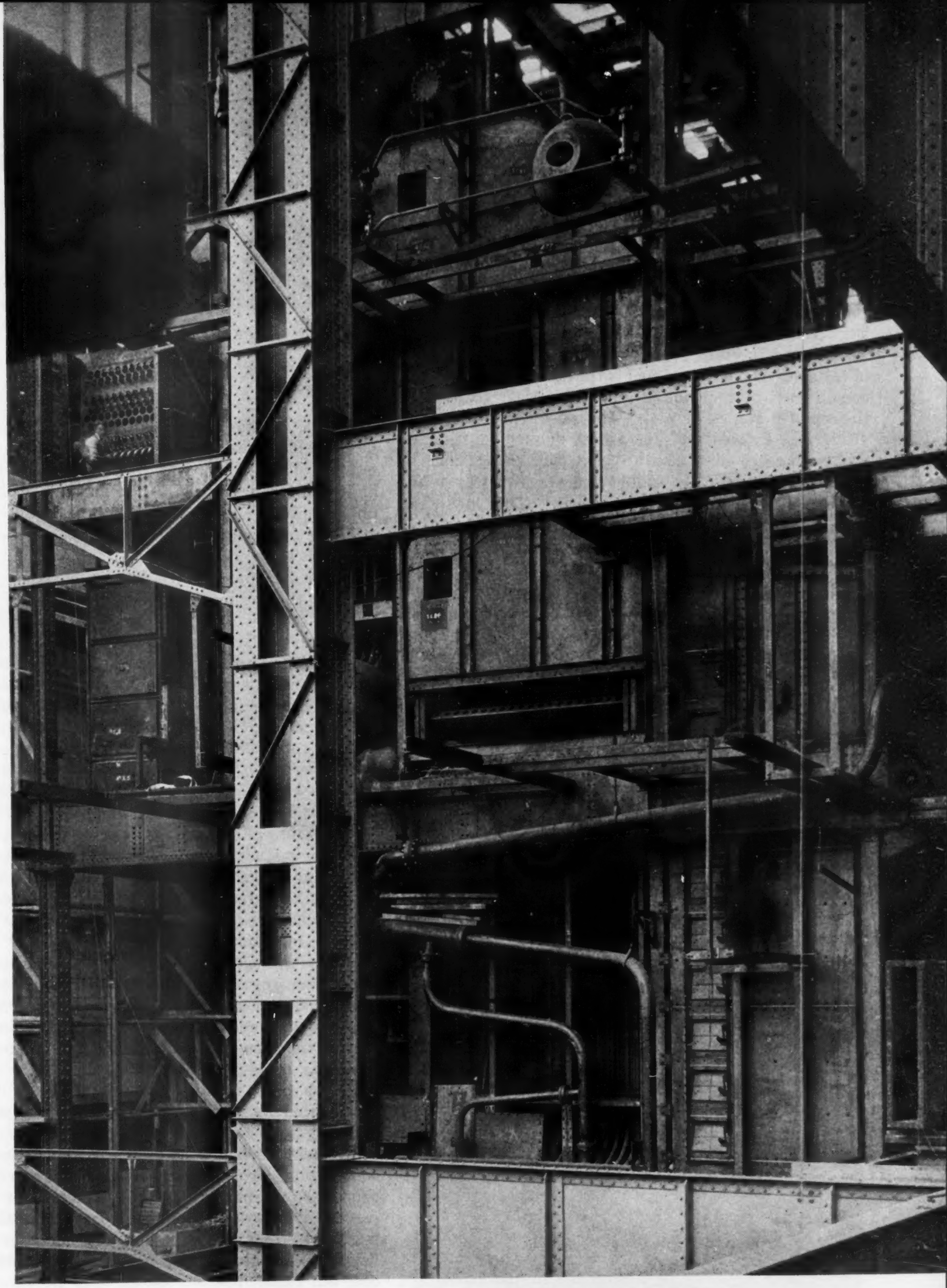
The first section of Tir John power station, which is due to go into full commercial operation before the end of the current year, comprises 120,000 kw of electrical generating equipment and four boilers of 280,000 lb per hr maximum evaporation. Leading particulars of the steam generation equipment, which is being constructed by International Combustion Limited, London, are as follows:

Number of boilers	Four
Type	Vertical bent tube with Lopulco fin-tube furnaces
Normal capacity	240,000 lb per hour
Overload capacity	280,000 lb per hour
Minimum guaranteed capacity	60,000 lb per hour
Design pressure	800 lb per sq in.
Working pressure	625 lb per sq in.
Final steam temperature	850 F.
Feed inlet temperature	350 F.
Maximum air temperature	700 F.
Guaranteed overall gross efficiency, based on gross califoric value	87.43 per cent

So far as general features are concerned the most interesting are the adoption of the "bin and feeder" system with provision for drying in the mill, bare tube water walls, long flame burners, highly preheated air and a flue gas washing system. These recommendations follow on general experience of pulverized fuel firing and may be taken as representative of the practice which has proved most satisfactory in each department. It is in the details of fuel preparation and firing, furnace design, etc., where the specialized data obtained with pulverized anthracite duff are to be observed.

Characteristics of Welsh Anthracite

One of the peculiarities of British fuels is the inconsistency of characteristics of coals even from the same coal field, and this has been found to apply to anthracites no less than to semi-bituminous coals. The Department of Scientific and Industrial Research has carried out very important investigations into the

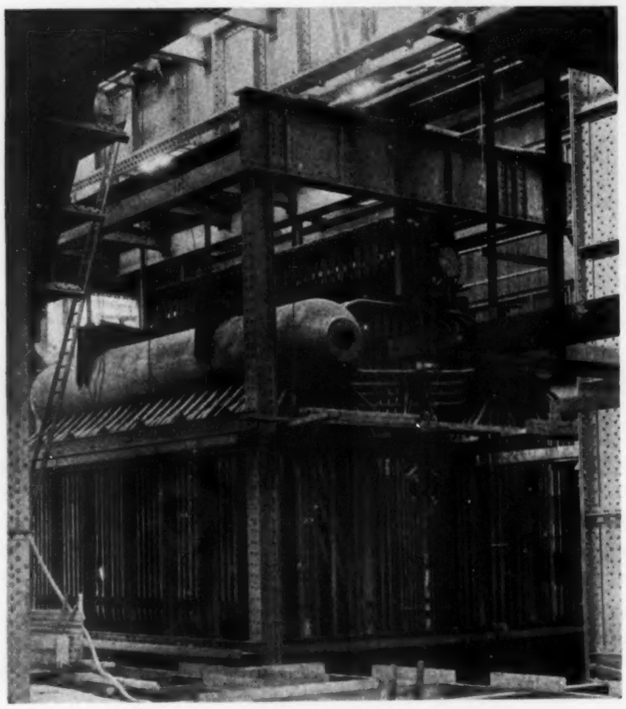


One of the four 240,000 lb per hr steam generating units at the Tir John Station with economizer and air heater

nature of coal and its publications have thrown much light on this problem. A knowledge of the physical and chemical properties of the banded constituents of coals explains many of the difficulties experienced in burning low-grade fuels and enables the behavior of such coals to be anticipated when their source is known.

The study of the constitution of coal, published in the form of a monograph, by Stopes and Wheeler, enables the wide discrepancy in the nature of apparently similar fuels to be partially explained. Dr. Lessing's work also has had invaluable practical applications. The most important revelation from the point of view of behavior as pulverized fuel is the nature of the banded constituents of coal and in particular their relation to the smaller grades of fuel. The above mentioned investigators showed that the lowest quality constituent, though present in comparatively small proportions in the seam, is extremely friable and consequently tends to concentrate in dusts, slurries, etc. This low-grade constituent has a relatively high inherent ash content which moreover has a low fusing temperature.

One of the most remarkable features of the low-grade band is the poor calorific value in comparison with the higher bands even though there may not be any appreciable difference in the proximate analyses. This can only be accounted for by assuming that the volatile constituents are of a very low order. This theory is supported by the fact that at temperatures in the region of 1,000 F the low-grade band takes twice as long to ignite as a more representative sample from the seam. The combustion of aspirated anthracite dust from coal cleaning plants has been rather closely investigated in South Wales and owing to the concentration of the low-grade constituent the conclusion was generally held at one time that this was not a practical proposition. After much experiment, however, this class of fuel has been successfully burned and, curiously enough, the results were obtained in a furnace with a large proportion of bare water-cooled surface in the walls.



Boiler unit in course of erection

The following analyses are typical of three types of anthracite duff obtained from collieries within a few miles of each other. These are all representative of fuels being burned at high efficiency under pulverized-fuel-fired boilers and also serve to illustrate the foregoing remarks:

Sample No.	1	2	3
Volatiles	12.05	5.68	8.75
Fixed carbon	72.39	75.39	71.80
Ash	13.80	16.60	18.20
Moisture	1.76	2.33	1.25
Gross calorific value	12,371	12,618	11,696
Gross calorific value on ash free and dry basis		15,518	14,518
Grading			
Through 40 mesh, per cent		22.75	99.95
Through 100 mesh		4.47	96.59
Through 200 mesh		1.44	82.20

Sample 1 was a banked duff, all through $^1/_8$-in. mesh, of indefinite age and burned easily with a turbulent natural draft burner, requiring little effort to establish and maintain ignition. When handled by a Raymond impact mill the life of the hammers was over 2,000 tons grinding to 75 per cent minus 200 mesh, before it was necessary to re-arrange them.

Samples 2 and 3 were from the same colliery but whereas the former, which was taken from the rejects from a screening plant, could be ground and burned reasonably well by means of long flame burners in a well cooled furnace with cold air, the third sample, an aspirated dust from the colliery dry cleaning plant, could be ignited only with the greatest difficulty and the flame remained unstable until the furnace had become thoroughly warmed up.

Sample 3 well illustrates the peculiar characteristics of the low-grade band for, although the volatile content is slightly higher than in the more representative Sample 2, the ash free and dry calorific value is 1,000 Btu lower per pound.

The last two samples are typical of the fuels regularly supplied to furnaces with bare water-cooled side and rear walls, using straight-shot burners. The walls gave no assistance when lighting up and very little when the furnace had warmed through. Consequently, apart from re-radiation from the front refractory wall, the success of this installation depended entirely on fuel and air regulation and considerable data accrued during the period while combustion of the aspirated dust was being perfected.

It has been found that anthracites of the nature represented by Samples 2 and 3 require very careful handling to secure efficient combustion.

The main difficulty is associated with the supply of combustion air and it has been concluded that the proportion of primary or carrier air must be closely regulated in order to ensure a coal-air ratio approximating to that requisite for the maximum speed of flame propagation. This is logical with any class of coal but experience has shown it to be almost vital in the case of low volatile coals with ignition temperatures of the order of 750 F.

Having established ignition, the balance of air for combustion needs to be admitted to the furnace under full control of quantity, direction and velocity until the flame is firmly established, in order to avoid excessive cooling to which a pulverized anthracite flame is extremely susceptible.

Since the small volatile content of anthracite is composed mainly of the heavier hydro-carbons it is not

readily evolved and ignition is consequently dependent almost entirely on heat absorbed from external sources.

A substantial lighting up equipment is consequently essential for starting up a cold boiler and radiant refractory is also necessary in the ignition zone and for such portion of the furnace as the flame traverses before the fuel has reached a condition of stable combustion. After this stage the fuel behaves more or less like any other coal after the volatiles have been liberated.

The fuel specified for Tir John Station is as follows:

Volatiles	6–8 per cent
Fixed carbon	74–80 per cent
Ash	14 per cent
Moisture	4 per cent maximum
Gross calorific value as fired	12,500 Btu per lb
Grading of raw coal:	
Through 50 I.M.M.	30 per cent
Through 20 I.M.M.	55 per cent
Through 10 I.M.M.	83 per cent
Through 1/8 I.M.M.	95 per cent

This is indicative of duff, and consequently should not present the difficulties experienced with the aspirated dust, since the volatile content should have normal constituents. It is not, however, so free burning as Sample 1, previously referred to, but the combined experience from all three classes of low-volatile coals underlies the design of the new station and will be available to ensure that the anticipated results are achieved.

Special Features of Tir John Boiler Plant

Although it has been found that anthracite is not necessarily as highly abrasive as once thought, Hardinge conical ball mills are installed as a safeguard against excessive maintenance. Four 10-ton mills are included on coal up to 2 per cent moisture without drying, grinding to 85 per cent minus 200-mesh I.M.M. Provision is being made for drying in the mill, suitable connections being arranged on the mill circuits, forced-draft ducts and furnaces for hot air supply and ventage disposal through bus-mains which are necessary to maintain the independence of mills and boilers. Only the ventage system to the furnaces is being installed with the initial plant.

An interesting feature of the fuel preparation equipment is the employment of pneumatic transport pumps between the cyclone collectors and overhead storage bunkers to reduce headroom. The pumping system is interconnected and under full control from a central panel.

The burners, twelve per boiler, are of the Lopulco straight-shot vertical type with tertiary air, incorporating minor modifications as the result of experience on the experimental plants previously mentioned. The burner is so simple that no radical alteration in design is involved. The chief departure from recognized practice is in the system of air supply and distribution.

Experience has shown the necessity of maintaining a temperature as near as possible to ignition temperature in the path of the fuel stream until the flame is well established. The combustion air will therefore be supplied to the primary air fan and tertiary air ports at 700 F and the secondary air will enter the combustion chamber at an even higher temperature when the furnace walls have warmed up. This will virtually preclude loss of ignition or delayed combustion, even allowing for the large water-cooled surface in the furnace.

In order to attain this high air temperature a unique combination of economizer and air heater has to be adopted, the air heater being divided into two sections in series with an interposed economizer. By this means an economic temperature head is maintained between flue gases and cooling medium from boiler exit to chimney. The pre-economizer air heater is fitted with a gas bypass with the object of limiting air temperature to 700 F and also reducing the load on the induced-draft fans at overload rating.

The secondary air distribution system through the furnace front wall will be divided into three independent vertical zones as well as the usual Lopulco system of horizontal passages. An arrangement of dampers will not only regulate the air supply during the downward sweep of the flame but will permit concentration of air in the central zone when the wing burners are out of service at low ratings.

Based on experience already obtained, it is anticipated that this combination of highly preheated air and close control of admission will ensure stable and efficient combustion under all conditions. The side and rear furnace walls are therefore protected by fin tubes up to burner arch level above which spaced bare tubes protect the refractories. The burner arch is also provided with water-tube protection and the hopper is spanned by a water screen.

In view of the high air temperature the Lopulco duplex feeders have been modified slightly in order to avoid the use of a perishable seal for the bearing of the vertical spindle.

In designing the furnace and boiler regard has been paid to the low fusibility of the ash. The ashpit is spanned by a double water screen. The front bank of boiler tubes has been set in such a manner as to permit the introduction of a convection superheater of the inter-deck type of sufficient heating surface to give the high degree of superheat specified. Ample gas passages are thus provided, readily cleared by soot blowers, and the vertical layout of the superheater avoids the use of supports on which flue dust can lodge and fuse.

In view of the use of straight-shot burners the furnace rating is comparatively low in comparison with the figures ruling with turbulent burners of the self-contained or tangential types, but the ability to protect the rear portion of the furnace with bare water tubes without detriment to combustion, once established, has permitted the adoption of a higher rating than might have been expected. This approaches 25,000 Btu per cubic foot per hour at maximum load.

The furnaces will be equipped with horizontal oil burners for starting, withdrawable through the side walls, and the oil burning system will be of ample capacity to ensure stable combustion before the oil burners are extinguished.

It is anticipated that the new station will be available for commercial load in November of this year, and operating results should be forthcoming shortly afterward.

The completion of this project is a matter of high importance not only to the combustion engineer, but as a very important further development in pulverized fuel firing practice.

Successful operation of a plant of this magnitude will no doubt lead to further extensive small scale applications of pulverized fuel firing, as a means of solving simultaneously the problems of cheap production and disposal of colliery waste.

Discharge Capacity of Traps

By A. E. KITTREDGE and E. S. DOUGHERTY

The Cochrane Corporation, Philadelphia, Pa.

While the theoretical considerations governing the flow of hot water through a nozzle are available in any text on thermodynamics, apparently little has been done to apply this knowledge to the problem of determining the maximum capacity of steam traps and the accurate prediction of capacities under varying conditions of pressure and temperature. In fact, a survey of the available information on trap capacities discloses that in many instances the claimed capacities are far in excess of the theoretical maximums possible with discharge orifices of the specified sizes and further that the usual method of trap size selection does not appear to treat the factors influencing trap capacities with their proper significance.

FREQUENTLY the maximum capacity required of a trap is unknown and selection is on the basis of individual judgment and previous experience modified by a generous factor of safety. This might be said to be particularly true of traps installed for emergency service, such as those draining steam lines and boiler steam purifiers. It is not true of traps serving equipment such as evaporators and closed feedwater heaters designed for a certain definite rate of steam condensation at a specified pressure. In the latter case exact knowledge of trap capacity is imperative.

From a survey of competitive information the capacity per square inch of port area indicated by Curve A, Fig. 1 was found to be representative of the capacities

Fig. 1—Curve of theoretical discharge compared with curve of guaranteed capacities

guaranteed by all manufacturers from whom such information was available. Curve B was prepared on the momentary assumption that in the equation

$$Q = \frac{A\sqrt{2gh}}{V} \qquad (1)$$

where, A = area
V = specific volume
h = head

the correct results are obtained by taking h equivalent to the total-pressure difference on the trap and by assuming that the specific volume of the fluid discharged is the specific volume of the liquid in its initial state. On this basis the trap capacities as stated appear to have a liberal margin of safety. The fallacy lies in the fact that the effective pressure difference is not the total pressure difference as assumed and the specific volume of the mixture passing through the valve seat is not the specific volume of the liquid in its initial state.

In considering means of accurately determining the maximum theoretical discharge of water at steam temperature through a nozzle, the first and most obvious conclusion is that the potential energy represented by the equivalent static head h is, at least in part, converted into kinetic energy in the form of velocity of the fluid discharged. It is equally apparent that as the potential energy is converted into kinetic energy and the static pressure is reduced, part of the water which was initially at steam temperature flashes into steam and greatly reduces the density of the fluid, it now consisting of a mixture of water and steam. Steam is generated whenever the total pressure of the water is reduced to a value less than the vapor pressure corresponding to the temperature of the water. It is believed that these facts

have been generally recognized by engineers concerned with the discharge capacity of steam traps but that lack of appreciation of their tremendous influence has been obscured by the absence of such an analysis of the problem as is here presented.

The obvious fact that steam is formed in a nozzle passing water which was at steam temperature at the initial pressure, suggests the probability that hot water has a critical pressure similar to the critical pressure of a gas or a vapor which greatly limits the discharge capacity of trap orifices and makes the maximum capacity a function of the absolute upstream pressure and upstream temperature whenever the discharge pressure is below this assumed critical point.

Appreciating that Equation (1) has been very generally misapplied to the problem of determining the maximum possible flow of high-pressure water at steam temperature and further that the conditions of flow, involving as they must the formation of steam, very likely create a critical pressure, the logical procedure is as follows: First, to determine the correct formula for flow under the existing conditions and to make a series of calculations of the flow of hot water from any pressure P_1 to several lower discharge pressures P_2, covering the range between P_1 and zero, for the purpose of determining the existence and locus of the critical pressure and the maximum flow from the initial pressure P_1, which, of course, occurs at the critical pressure.

From the law of conservation of energy, the kinetic energy represented by the rate of discharge of any fluid from a nozzle is equal to the loss of intrinsic energy of the fluid discharged plus the loss of potential energy of the fluid discharged. Therefore, the fundamental equation applicable in this case is

$$\frac{V^2}{2g} = 778(u_1 - u_2) + 144(P_1V_1 - P_2V_2) \qquad (2)$$

where V = velocity ft/sec
g = acceleration due to gravity
u_1 = initial intrinsic energy, Btu/lb
u_2 = final intrinsic energy, Btu/lb
P_1 = initial pressure lb/sq in.
P_2 = final pressure lb/sq in.
V_1 = initial volume, cu ft/lb
V_2 = final volume, cu ft/lb

In the case of discharging water at a temperature high enough to cause flashing in the process of discharge, the

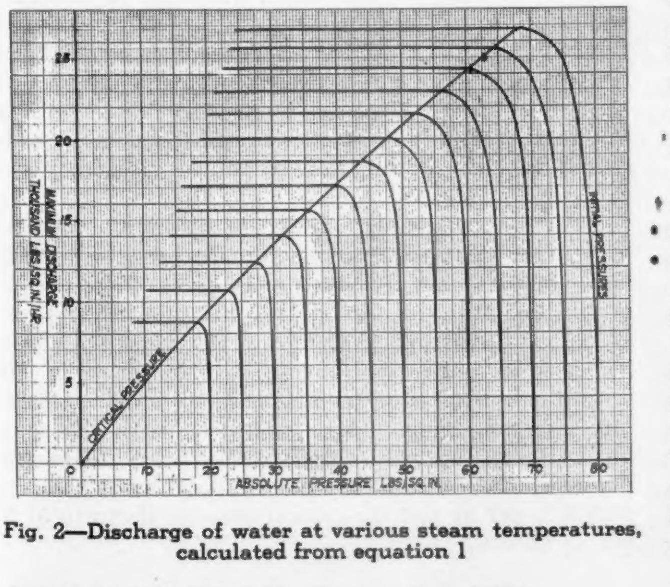


Fig. 2—Discharge of water at various steam temperatures, calculated from equation 1

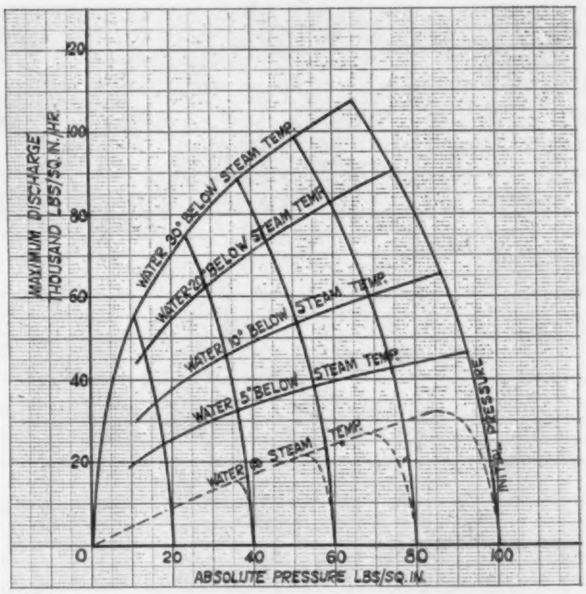


Fig. 3—Maximum discharge versus absolute pressure (up to 100 lb per sq in.) for various differences between water and steam temperature

initial intrinsic energy u_1 is the liquid heat of the water at the existing temperature and the final intrinsic energy u_2 is the liquid heat remaining at pressure P_2 plus the product of the quality of the steam found existing at P_2 and the heat equivalent of internal work contained in a unit of weight of steam existing at pressure P_2.

Equation (2) as stated involves all the factors which influence the maximum rate of discharge of a flashing liquid from any pressure P_1 to any pressure P_2. It is discussed for its theoretical value but is not recommended for use in actual calculations of the flow of hot water due to the unavoidable defect of this method of calculation of external work during expansion, in that it depends upon taking the difference of quantities which are of the same order of magnitude. Equation (1) is equally correct, theoretically, when the proper substitutions are made for head and specific volume and is a much more accurate procedure.

To apply correctly Equation (1) to the problem of the discharge of hot flashing water the specific volume of the mixture at P_2 must be determined and the pressure difference $P_1 - P_2$ converted into equivalent feet h of the mixture at P_2. Similarly, the specific volume of the mixture discharged is, of course, that at P_2 on which the head h is based. Correct determination of the specific volume V_2 at pressure P_2 involves, first, finding the heat available between the liquid heat existing at P_1 and the liquid heat corresponding to any pressure P_2 with which we are concerned. The percentage of steam formed is the heat drop per pound between q_1 and q_2 divided by the latent heat at P_2. This is not absolutely correct since it assumes the total heat content at P_2 to equal the total heat content at P_1. Actually, the steam formed by the process of flashing expands adiabatically from the pressure at which it is formed to P_2. Omission of this correction factor will not affect the final results more than 1 per cent, although it has been included in the calculations to arrive at the results plotted herein. The specific volume of the mixture is, therefore, the specific volume of liquid at P_2 plus the product of the quality at P_2 and the specific volume of evaporation at P_2.

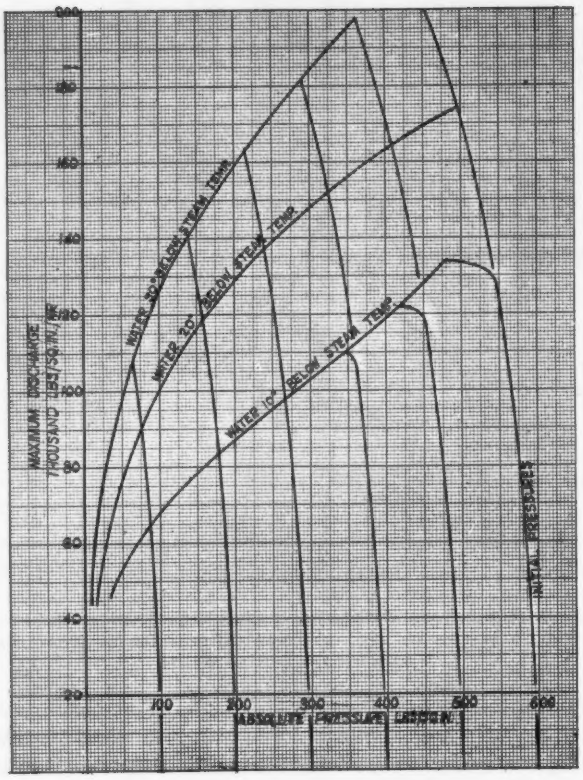


Fig. 4—Maximum discharge versus absolute pressure (up to 600 lb per sq in.) for various differences between water and steam temperature

The equivalent feet h of the mixture for insertion in Equation (1) is therefore $(P_1 - P_2)V_m \times 144$ where V_m is the specific volume of the mixture at P_2.

Now, by applying Equation (1) to the problem of determining the rate of discharge Q of water at steam temperature, P_1 initial pressure, through unit area A to any discharge pressure P_x below P_1, it will be found that as P_x is reduced the value Q will increase with a decreasing rate until a maximum value is reached at some intermediate point between P_1 and zero pressure and will then decrease with an increasing rate, approaching zero as P_x approaches zero. The value of P_x at which Q is a maximum, is known as the critical pressure and the values of P_x below the critical pressure so determined are hypothetical. That is to say, the values of P_x below the critical were assumed to exist as a means of plotting the discharge curve to obtain the critical pressure but do not actually exist in the throat of a nozzle. Similarly, the rate of discharge Q for discharge pressures below the critical does not decrease but maintains a constant value equal to that existing at the critical pressure.

The critical pressure thus determined is exactly analagous to the critical pressure of a gas or a vapor such as air or steam and exists by virtue of the fact that at that pressure the ratio of velocity resulting from pressure drop to specific volume of the mixture is the greatest possible.

Fig. 2 illustrates a family of curves plotted from values calculated by use of Equation (1) as correctly interpreted above. Each individual curve, rising vertically from the abscissa, is plotted for that initial pressure indicated by the terminus of the curve with the abscissa. The maximum discharge in thousand pounds per square inch per hour is found by reading the ordinate value directly to the left of the intersection of the curve indicating initial pressure with the discharge pressure P_x read vertically from the abscissa. The diagonal curve passing through the origin of the chart and through the apices of the discharge curves indicates the critical discharge pressure.

Reading from Fig. 2, we find that the maximum theoretical discharge from an initial pressure of 80 lb per sq in. to a discharge pressure of $77^1/_2$ lb per sq in. is 20,000 lb per sq in. per hr. Similarly, we find that the maximum theoretical discharge from an initial pressure of 80 lb per sq in. to any pressure lower than 68 lb abs is 26,800 lb per hr.

Curves similar to those shown in Fig. 2 were plotted for initial values of 100, 200, 300, 400, 600, 900 and 1200 lb per sq in. absolute pressure and from these curves maximum discharge rates and critical pressures were observed. Since most problems of trap discharge concern discharge pressures below the critical pressure so determined, the character of the discharge curve between the initial pressure and the critical pressure is not extremely important and to plot accurately such curves requires a longer abscissa scale than used in Fig. 2, which is itself presented to illustrate the method of procedure.

Influence of Temperature Depression

The curves plotted in Fig. 2 indicate only the discharge of water at steam temperature. From the information presented above it is apparent that whenever the discharge pressure P_2 to which any trap or nozzle passing a liquid may discharge falls below the vapor pressure of that liquid, as indicated by its temperature, the volume of the mixture at the discharge of the trap or nozzle will be influenced and a critical pressure will exist. It is further apparent that as the vapor pressure of the liquid approaches the discharge pressure P_2 at the outlet of the trap or nozzle the critical pressure P_c will approach P_2 and entirely disappear when the vapor pressure of the liquid is less than the discharge pressure P_2.

Since the factors controlling maximum discharge and critical pressure for water at less than steam temperature are the same as for water at steam temperature, exactly the same procedure may be followed to determine these values. Accordingly, there are plotted in Figs. 3 and 4 curves showing the maximum discharge in thousand pounds per square inch per hour of water at various temperature depressions in degrees Fahrenheit below steam temperature. The curves shown in Figs. 3 and 4 are to be used in exactly the same manner as those shown in Fig. 2 and for purpose of comparison Fig. 3 includes in dotted lines the information contained in more detail in Fig. 2.

Referring to Fig. 3 we find that with an initial pressure of 80 lb per sq in. abs the maximum discharge of water at 5 deg below steam temperature is 42,500 lb per sq in. per hr, and that the critical pressure is 74 lb per sq in. absolute as compared to the respective values of 26,800 lb per sq in. per hr discharge and 67 lb per sq in. abs critical pressure for water at the same initial pressure but at steam temperature. Similarly, with the same initial pressure but with water at 30 deg below steam temperature, the maximum theoretical discharge rises to 98,500 lb per hr and the critical pressure drops to 51 lb per sq in. abs.

It appears worth while to observe that, as illustrated by Fig. 2, the discharge curves for water at steam temperature approach the critical pressure gradually, having tangents at the critical pressure parallel to the horizontal cross-section lines, whereas the capacity curves, shown in Fig. 3, terminate abruptly at the critical pressure, as indicated by the diagonal curve denoting temperature depression. This change in the character of the curves is caused by the fact that in the case of water at steam temperature flashing begins simultaneously with acceleration of the fluid and has a gradual influence on the discharge capacity, whereas in the case of water at less than steam temperature flashing begins after a great deal of acceleration has taken place. The result is that in the latter case the critical pressure is frequently fixed almost exactly by the vapor pressure of the liquid. Actually, the discharge curve shown in Fig. 3 must also have horizontal tangents at the critical pressure but the critical pressures were found to be so close to the vapor pressure of the liquid that the two were indistinguishable.

Notes on Critical Pressures

The curves shown in Figs. 2, 3 and 4 indicate critical pressures in pounds per square inch absolute for any initial pressure for which a discharge curve is drawn. Referring to Fig. 3, it will be noted that for an initial pressure of 80 lb per sq in. abs the critical pressure with 5 deg temperature depression is higher than the critical pressure with water at steam temperature, whereas the critical pressure with 20 or 30 deg temperature depression is lower than the critical pressure of water at steam temperature and the critical pressure with 10 deg temperature depression approximately coincides with the latter. To provide a graphical picture of the variation of critical pressure with variation of initial pressure and temperature depression the values of critical pressure taken from Figs. 2, 3 and 4 have been plotted as a percentage of the initial pressure against the initial pressure. This is shown in Fig. 5. It is interesting to note from Curve 1, Fig 5 that the critical pressure of water at steam temperature approaches the initial pressure as the initial pressure approaches zero and decreases in a smooth curve to the critical pressure of steam at 3226 lb per sq in. abs at which pressure the characteristics of water and steam are the same.

Simplified Curves for Ordinary Use

Since from inspection of Fig. 5 it will readily be agreed that the great majority of engineering problems involving the discharge of hot water concern the discharge to pressures below the critical pressure, the information on discharge rates contained in Figs. 2, 3 and 4 may be replotted in the more simple form shown in Figs. 6 and 7 to indicate maximum discharge rates from any initial pressure P_1 to any discharge pressure P_2 below the critical for water at steam temperature or at various degrees Fahrenheit below steam temperature. These curves assume a back pressure at the valve seat of the trap not in excess of the critical pressure indicated for the same initial pressure and temperature conditions as in Fig. 5. The curve denoting discharge rates of cold water and saturated steam in Fig. 6 are included for the purpose of comparison and in the case of cold water, the vapor pressure of which approaches zero, the abscissa may be interpreted as differential pressure.

All of the discharge rates given in Figs. 2, 3, 4, 6 and 7 are theoretical maximums per square inch of port area and must be multiplied by suitable discharge coefficients to find actual trap discharge capacities. From the author's analysis of a capacity test of a steam trap having a valve port area of one square inch conducted under operating conditions with the temperature of the water entering the trap carefully recorded, a flow coefficient greater than 50 per cent is very improbable in a high capacity trap of the piston-operated or balance-valve type. This coefficient is applicable to the capacity ratings given in Figs. 6 and 7 whenever the absolute back pressure at the trap does not exceed the critical pressure as indicated for that initial condition of pressure and temperature by Fig. 5.

Actually, the tests in question were run with several different back pressures at the trap but in each instance where the absolute back pressure was less than the critical pressure the ratio of the actual weighed discharge to the theoretical maximum predicted by use of the data contained in Figs. 6 and 7 was constant.

The discharge coefficient applicable to bucket-type steam traps, having discharge orifices relatively much smaller in comparison with the cored passages through the trap than usually obtain in the case of high capacity balanced or piston operated traps, should be somewhat more favorable than the value of 50 per cent found by test to hold in the latter case and the writer suggests a maximum coefficient of 60 per cent for bucket steam traps.

An interesting comparison can now be made between Curve A, Fig. 1, representing the guaranteed capacity of a bucket trap of one manufacturer converted for uniformity into discharge per square inch per hour, and those curves in Figs. 6 and 7 indicating the maximum theoretical discharge of water at steam temperature. From Curve A, Fig. 1 we find the guaranteed capacity of 85,000 lb per sq in. per hr, when operating under a "pressure difference" of 100 lb per sq in. Assuming this rating to apply to discharge from 100 lb gage to atmosphere, we find by referring to the curve for discharge capacity of water at steam temperature in Fig. 7 that the maximum theoretical discharge rate from 100 lb gage pressure (115 lb abs) is 35,000 lb per sq in. per hr. Multiplying this rating by a discharge coefficient of 60 per cent indicates a maximum actual discharge of 21,000 lb per hr as compared to the claimed capacity of 85,000 lb per sq in. per hr. The values indicated by Curve A, Fig. 1 are said to be corrected for the temperature of the water and therefore are also assumed to apply to water at

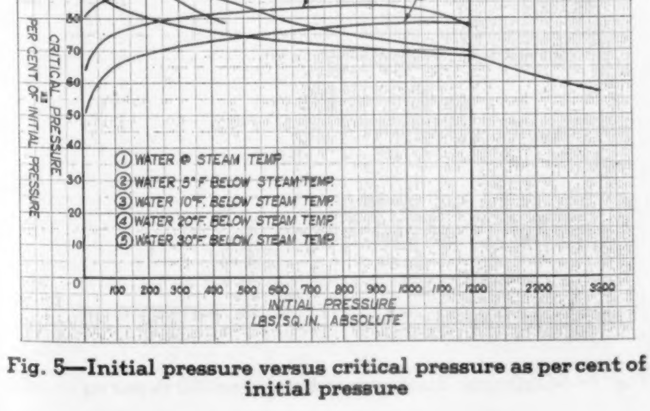


Fig. 5—Initial pressure versus critical pressure as per cent of initial pressure

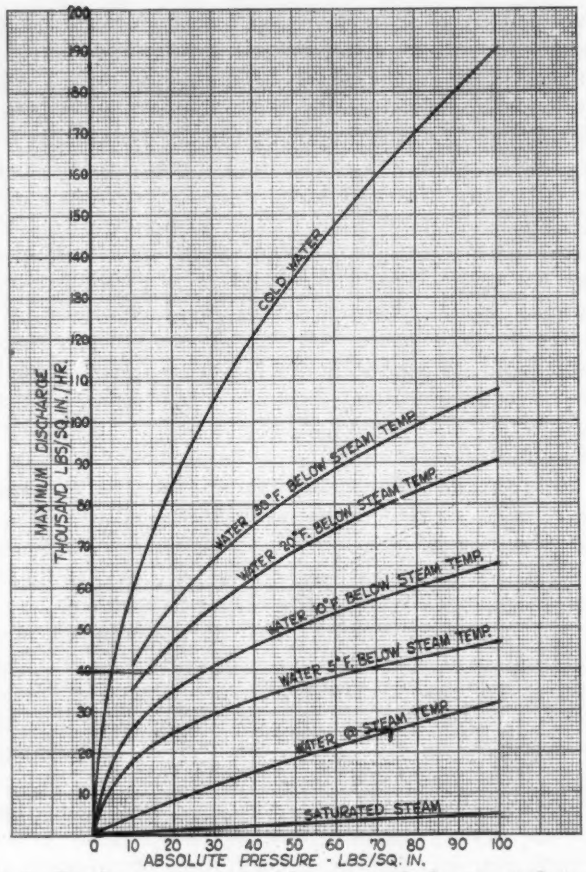


Fig. 6—Discharge rates of cold water and of saturated steam

steam temperature. This is not an isolated or even exceptional case but is rather the most logical and consistent of information available from three manufacturers.

Perhaps the most important disclosure made by this analysis is the fact that the discharge capacity of a steam trap positively cannot be predicted from knowledge of the differential pressure across the trap alone and any manufacturer presenting tables or curves of steam trap capacities based on differential pressures through the trap is either poorly informed or deliberately disguising the true facts.

In view of the fact that this analysis of the problem of discharge capacities of steam traps conflicts with the usual method and challenges the guaranteed capacities of some manufacturers said to be based on tests, it is essential that the reader should also be advised that many of the values given in Figs. 5, 6 and 7 have been checked by tests of a special nozzle for this purpose.

Capacity of Steam Trap Discharge Pipes

A successful steam trap installation requires a discharge pipe, large enough to handle the specified capacity without creating a back pressure at the trap discharge greater than the critical pressure as given by Fig. 5. The most economical steam trap installation will have a discharge pipe no larger than that size required to handle the specified capacity without creating a back pressure at the trap discharge greater than the critical pressure as given by Fig. 5, since a larger pipe than this would add nothing to the trap capacity.

Obviously, the discharge pipe length, diameter and roughness, together with the density and viscosity of the fluid flowing in that pipe, are the factors controlling the pressure loss or back pressure due to friction. There is, however, another factor which substantially influences the total back pressure at the trap for any one set of conditions. That factor concerns the kinetic energy which must be acquired by the fluid flowing through the discharge pipe in order to discharge the specified amount from a discharge pipe of known diameter under the conditions existing in that pipe near the discharge end.

We have seen from the preceding discussion that any expansible fluid has a critical pressure occurring where the ratio of the velocity resulting from pressure drop to the specific volume of the mixture is a maximum. Since it is desirable from the standpoint of economy to use that size of discharge pipe which will maintain back pressure at the trap equal to the critical pressure as given by Fig. 5, and appreciating that the diameter of the discharge pipe required will normally be a great deal more than the diameter of the trap orifice, we can assume that the fluid entering the discharge pipe is in the condition indicated by the critical pressure at Fig. 5 and will have substantially zero kinetic energy. The latter assumption is not exactly true, in fact, based upon the critical pressure existing at the trap discharge and the pressure existing at the end of the discharge pipe it will be found that the initial kinetic energy is about 10 per cent of the total at the end of the discharge pipe, based on the most economical pipe size as defined above and discharging condensate entering the trap at steam temperature. However, the factor of roughness, irregularity and size of the cored passages in the trap beyond the trap orifice must be considered and altogether the assumption that the initial kinetic energy in the discharge pipe is zero is not thought to be too conservative.

Knowing the condition of the mixture at the discharge of the trap to be that existing at the critical pressure P_c, we can treat the problem of discharge pipe capacity ex-

Fig. 7—Maximum discharge rates up to 1200 lb per sq in. abs

actly as the problem of trap orifice capacity, calculating the theoretical discharge rate from P_c to several pressures below P_c to determine at what pressure below P_c the discharge rate, exclusive of pipe friction, is a maximum. From a series of such calculations the values for Curve 1, Fig. 8, were obtained. Curves 2 and 3, Fig. 8, indicate critical pressure at trap discharge and pressure in the discharge pipe at outlet end, respectively. The back pressure in the receiving vessel may be as high as that value indicated by Curve 3, Fig. 8, without influencing either the size of trap or size of discharge pipe required. Dimension A therefore indicates the maximum useful pressure difference on the trap and similarly dimension B indicates the required pressure difference on the discharge pipe when the most economical pipe size is used. Required trap capacity divided by pipe discharge rate, as per Fig. 8, is the net effective discharge pipe area required exclusive of pipe friction. This value is used as the ordinate of Fig. 9 and is there identified as "Theoretical Discharge Pipe Area—Square Inches."

The pressure difference B, Fig. 8, in terms of feet head of the mixture, at the condition indicated by Curve 3, is the sum of the friction head plus the velocity head at the end of the discharge pipe. Since the actual discharge of any pipe of any known diameter and length may be determined from test, the actual discharge velocity may be calculated from the test discharge rate after determining the specific volume of the mixture at the end of the discharge pipe from measurement of the pressure in the pipe near the discharge end. We may therefore equate head B to the sum of the velocity head at the end of the pipe plus the friction head over the length of the pipe, and from that equation determine the friction coefficient when the average density is known. This procedure is correct only when the correct average density of the fluid is known. However, if it can be shown that for all conditions with which we are concerned the density at the end of the discharge pipe has a fixed rela-

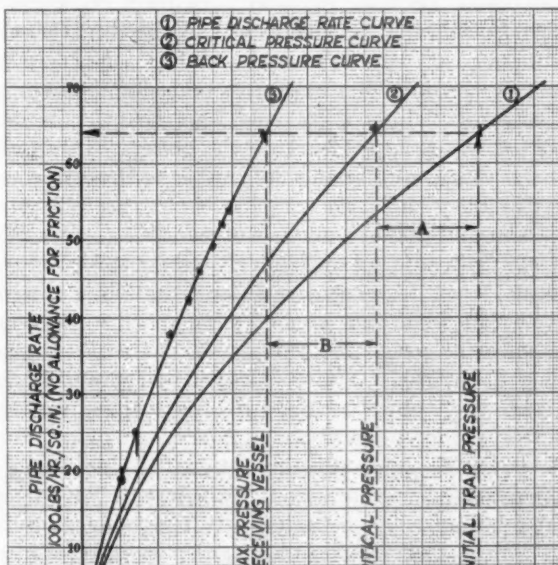


Fig. 8—Pipe discharge rate chart

Use Curve 1 to determine theoretical discharge rate per unit of pipe area as established by pressure drop "B" from critical pressure per Curve 2 to pressure not more than maximum permissible back pressure per Curve 3. To determine theoretical pipe area required divide required capacity by the theoretical pipe discharge rate. Use theoretical pipe area and equivalent pipe length in Chart, Fig. 9, to determine actual pipe size.

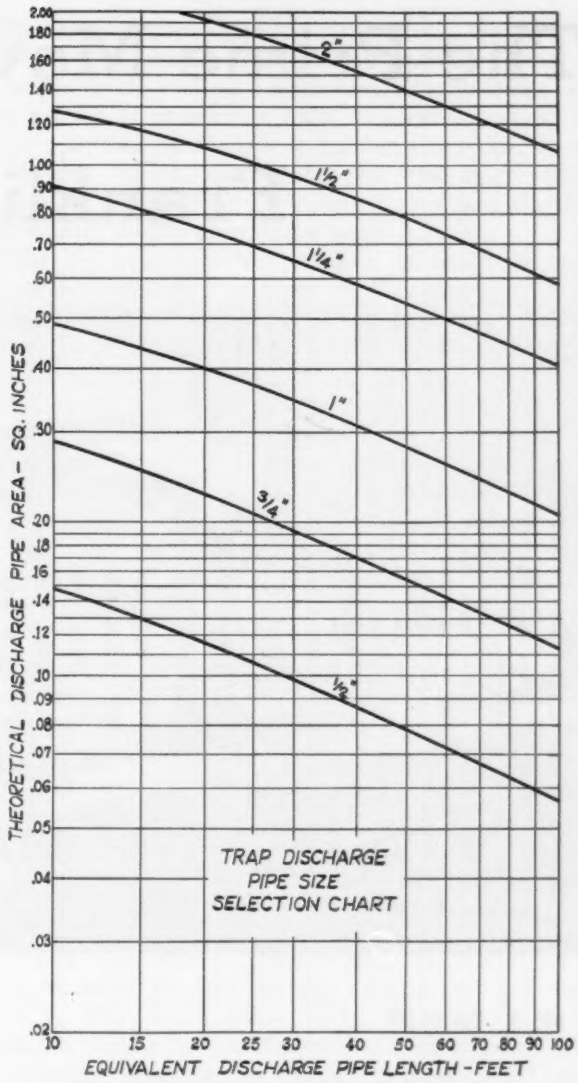
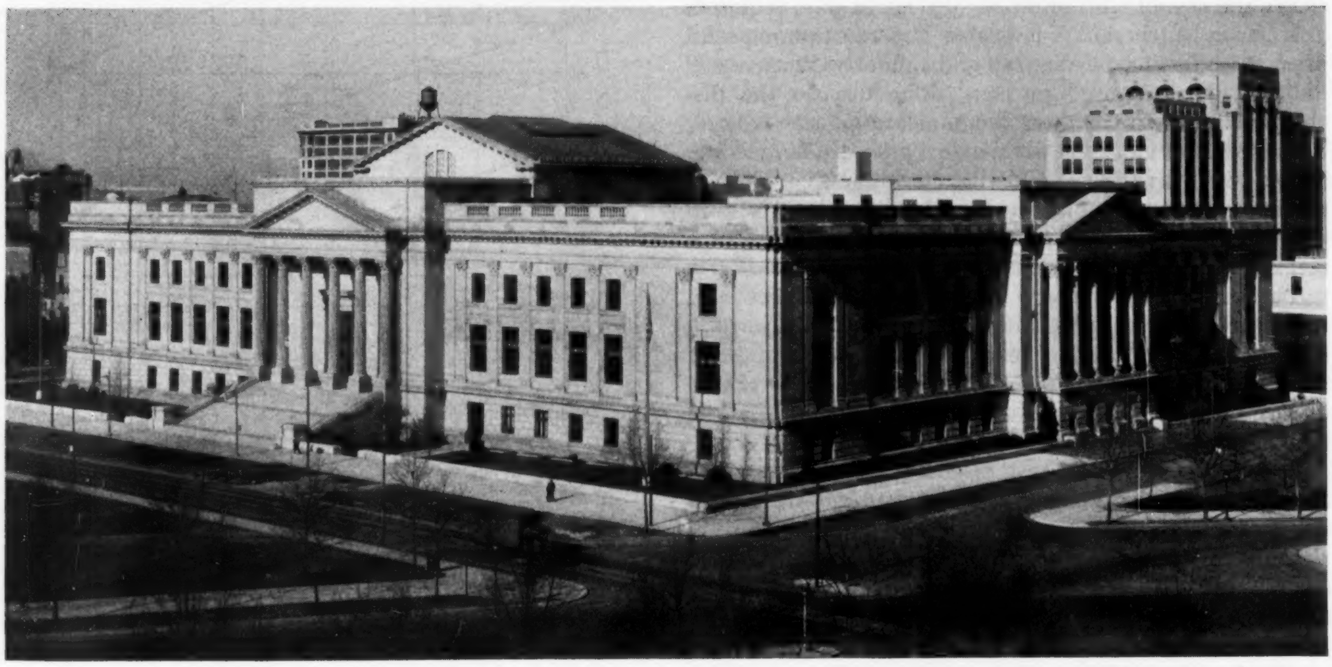


Fig. 9—Chart for determining discharge

tion to the average density then the friction coefficient may be determined and used in terms of the density at the end of the discharge pipe.

From an inspection of the steam tables it is found that, for water entering a trap at steam temperature under any pressure within the range shown by Curve 1, Fig. 8, the fluid density at the critical pressure indicated by Curve 2 has a substantially constant ratio to the fluid density at the pressure indicated by Curve 3. It is therefore reasonable to assume that the average fluid density in the discharge pipe has a substantially constant ratio to the fluid density at the outlet end of the pipe when the inlet and outlet discharge pipe pressures are as given by Curves 2 and 3 respectively, Fig. 8. Since, as previously explained, the latter conditions do exist at the inlet and outlet ends of a trap discharge pipe, when the pipe is loaded to its maximum capacity, and further since we are concerned only with factors affecting the discharge capacity of any pipe under these conditions, we may proceed to determine the friction coefficient in terms of the fluid density at the condition existing at the end of the pipe. The friction constant used for preparation of Fig. 9 was derived in this manner from testing a ½-in. pipe 10 ft long. Other points on Fig. 9 were calculated on the assumption that friction head varies directly as the product of density, pipe length and velocity squared and inversely as the inside pipe diameter.

The Prime Movers Exhibit at the Franklin Institute



Home of the Franklin Institute, Philadelphia

By H. S. LEWIS

THE prime movers exhibit at the Franklin Institute in Philadelphia is popular as indicated by the large crowds that have visited the museum daily since it has been open. It is worth an engineer's time to see it, as I will attempt to show.

The exhibit illustrates the refinements which through the years have been made in prime movers, including those actuated by steam, internal combustion, wind and hot air, water and muscle power. There are thirty operating exhibits to illustrate the mechanism and progress of the various classes.

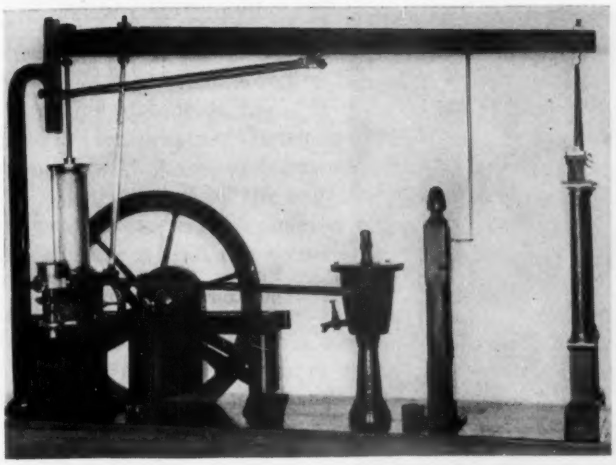
The earliest prime mover using steam is a working model of the walking-beam Newcomen atmospheric engine, of 1711, and used for pumping water. Steam is admitted at the base of a vertical cylinder under a piston. A spray of cold water injected into the cylinder condenses the steam and produces a partial vacuum, and the atmospheric pressure on the upper side of the piston forces it down. Then readmission of steam again forces the piston upward. Rope packing was used in place of piston rings, for it was not until 1816 that Barton's spring piston packing ring was invented. Tallow was generally used as the cylinder lubricant.

A working model of a 10-hp sun-and-planet engine invented by James Watt in 1782 illustrates the progress made in the 71 years. The first sun and planet engine for commercial purposes was built in 1785. It was a walking-beam engine with steam entering both top and bottom of a vertical cylinder. Instead of a sliding crosshead it had a parallel motion, but the throttle valve, centrifugal ball governor, air pump, double-acting pistons, steam-jacketed steam cylinder and condenser were, in principle, as of today. The steam after being condensed was pumped back to the boiler. The poppet valves for steam admission were similar to those now used on automobiles.

The sun-and-planet engine was constructed by Watt to circumvent patents held by others on the crank and connecting rod. A geared wheel on the shaft with the flywheel, called the sun, meshed with another on the end of a connecting rod, called the planet, the two being held in mesh by a link. The two gears being of the same diameter, the "sun" made two revolutions for each revolution of the "planet" about it.

Oliver Evans' high-pressure steam engine, built in 1817, was tested with steam at from 194 to 200 lb gage. It had a 20-in. steam cylinder, a 5-ft stroke, a rotative steam valve and four double-acting pump cylinders. Evans, a pioneer in high-pressure steam, saw its advantages, but lacked the necessary funds to continue his work.

Another model of the walking-beam engine is similar to the original engine built in 1842 for the Fairmount Water Works in Philadelphia. Also a walking-beam engine built in 1847 operated for 84 consecutive years. It was built with a centrifugal ball governor, and had a



Model of Oliver Evans high-pressure pumping engine with rotative steam valve built in 1817

15-in. bore and a 48-in. stroke. The boiler feed pump, air pump and feedwater heater were enclosed in the supporting column. It was constructed with an 8-in. diameter cast-iron shaft which was remarkably free from blow holes.

The Chambers engine built in 1849, which now operates with compressed air, was the smallest engine ever built to run on its own power. It had a $^1/_{16}$-in. steam cylinder by $^3/_{16}$-in. stroke and weighed $^1/_2$ ounce. Built of gold and silver by a 16-year old boy, it was displayed at the Centennial Exposition of 1876 in direct contrast with a great Corliss engine which at that time was the most powerful built.

In a Green vertical engine, built about 1876, the cylinder was placed above the crank and the piston rod entered at the top of a vertical cylinder. It was first used in the soda water industry.

A Corliss engine on display built in 1885 has two dashpots which help close the intake valve quickly. It operated for 45 years.

The Porter Allen engine, built in 1888 by the Southwark Foundry and Machine Co. was an early type of high-speed engine, operating at 275 rpm. The high speed was attained by effecting a balance of the moving parts. It had two distinct valves, one on the intake and one on the exhaust. It was constructed with a side crank and was used to drive some of the earliest d-c generators in Philadelphia.

A vertical compound launch engine with reversing gear built in 1900 and used in the Navy illustrates compound steam admission and the Stevenson reversible link motion.

A 100-hp Herrick rotary engine built in 1912 had a single blade double intermeshing rotor. It ran at 800 rpm and had a water rate of 32 lb per bhp per hr.

In a sectioned operating model of a simple engine the point of steam admission, cutoff, release and compression are illustrated. The engine is constructed with the Stephenson Link reversing gear which consists of two short cranks driving two concentric rods hinged to a link whose radius of curvature is equal to the length of the eccentric rods. The travel of the link from one extremity to the other gradually changes the angle of advance and controls the distance the valve moves. The valve receives its maximum travel at either extremity of the link and minimum travel at the center of the link. Thus, cutoff takes place early in the stroke.

The latest steam-driven equipment is the turbine. Though early models date back to 1825 the actual application for commercial developments began about 1880. The matter of priority is hard to establish, however; early practical development having been due to the efforts of DeLaval in Sweden for the single wheel, Parsons in England for the reaction type and Curtis in this country for the multi-stage type.

Two DeLaval impulse turbines, in which steam is completely expanded in a set of diverging nozzles and all the kinetic energy is given up to a single row of blades, are sectioned for illustration. Each turbine is equipped with a reduction gear in a 10 to 1 ratio. A 30-hp turbine built in 1902 and geared to a Sprague d-c generator, operated at 20,000 rpm and the generator speed was 2000 rpm. The 20-hp turbine built in 1904 ran at 24,000 rpm and had a pulley speed of 2400 rpm.

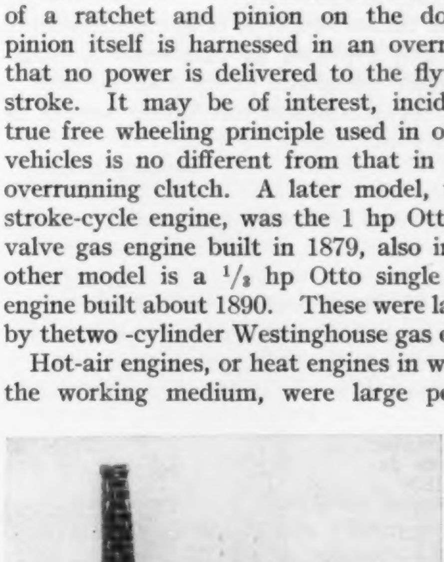
A sectioned 6-hp Kerr, two-stage, non-condensing turbine, built in 1906 ran at 1500 rpm. This turbine used the principle of the Pelton bucket. It operated until 1932 at which time it was replaced by a modern unit.

The sectioned 300-hp Parsons reaction turbine, where the energy of reaction from expansion in the moving blades on the rotor is added to the impulse of the steam as received from the fixed blades attached to the casing, ran at 3600 rpm. It was built in 1915 by the Allis-Chalmers Mfg. Co.

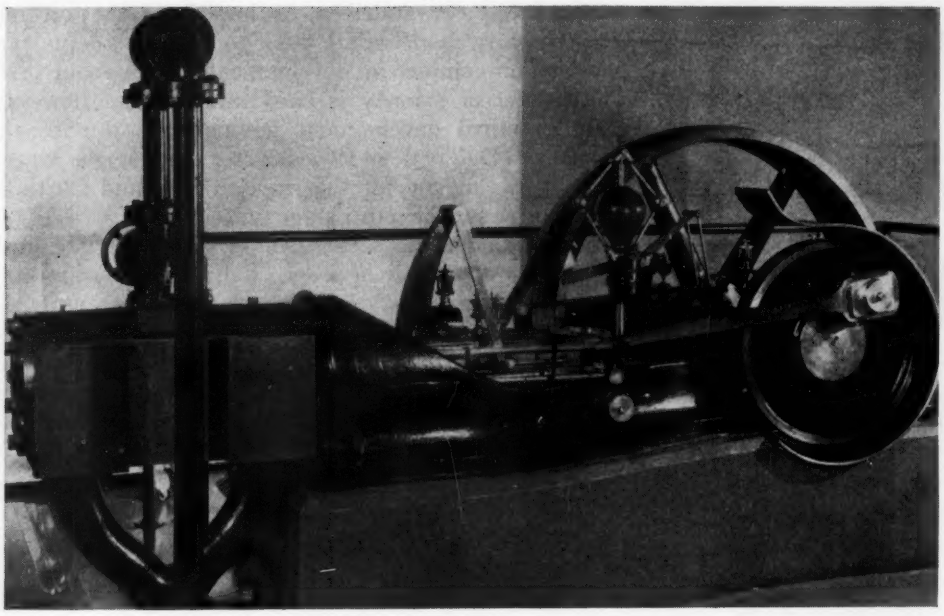
A sectioned Terry steam turbine illustrates an impulse turbine of the re-entry. type. It is an early model and, except for construction details, is similar to those in use today.

Internal-combustion engines are illustrated by an early model of the Otto Langen atmospheric gas engine built in 1867 in Germany. The operation is on the principle similar to the Newcomen atmospheric engine. In this case, however, the upward impulse to the piston is given by an explosion and the power stroke is the downward one. Power is stored in a flywheel by means of a ratchet and pinion on the down stroke. The pinion itself is harnessed in an overrunning clutch so that no power is delivered to the flywheel on the upstroke. It may be of interest, incidentally, that the true free wheeling principle used in our modern motor vehicles is no different from that in the Otto Langen overrunning clutch. A later model, the original four-stroke-cycle engine, was the 1 hp Otto horizontal slide valve gas engine built in 1879, also in Germany. Another model is a $^1/_3$ hp Otto single cylinder vertical engine built about 1890. These were later supplemented by thetwo -cylinder Westinghouse gas engine.

Hot-air engines, or heat engines in which air is used as the working medium, were large per horsepower of



Actual size of model of Chambers engine. This was built in 1849 by a 16-yr old boy and ran on its own power



Porter Allen high-speed engine used to drive early d-c generators in Philadelphia

capacity as compared with steam or gas engines. An Ericcson hot-air engine exhibited is an early specimen of the single-cylinder engine. A Rider Ericcson engine, built in 1870, is also exhibited. It shows the firepot, hot and cold cylinders, regenerator, cooling water pump and the water jacket. An operating fan using a high-speed hot-air engine illustrates the principles and one of the many applications of the engine.

A model of a flour mill built by the U.S. Wind & Engine Co. operated by wind, illustrates the application of wind power. A large fan geared to a shaft inside the mill is revolved by the wind. The speed of the fan is controlled by a governor which opens or closes small fins built into the blades of the fan.

A late model of the Corcoran windmill with the modern fixed blade wheel, operating a pump, has its speed controlled by a hand governor.

Water wheels are the earliest type of hydraulic motors and have been largely replaced by turbines. A model of an undershot breast wheel and pump, built in 1818 is the original, used for the Fairmount Water Works in Philadelphia. It was adapted for very low water heads, generally under six feet, and operates by water being directed against buckets on the periphery of a wheel.

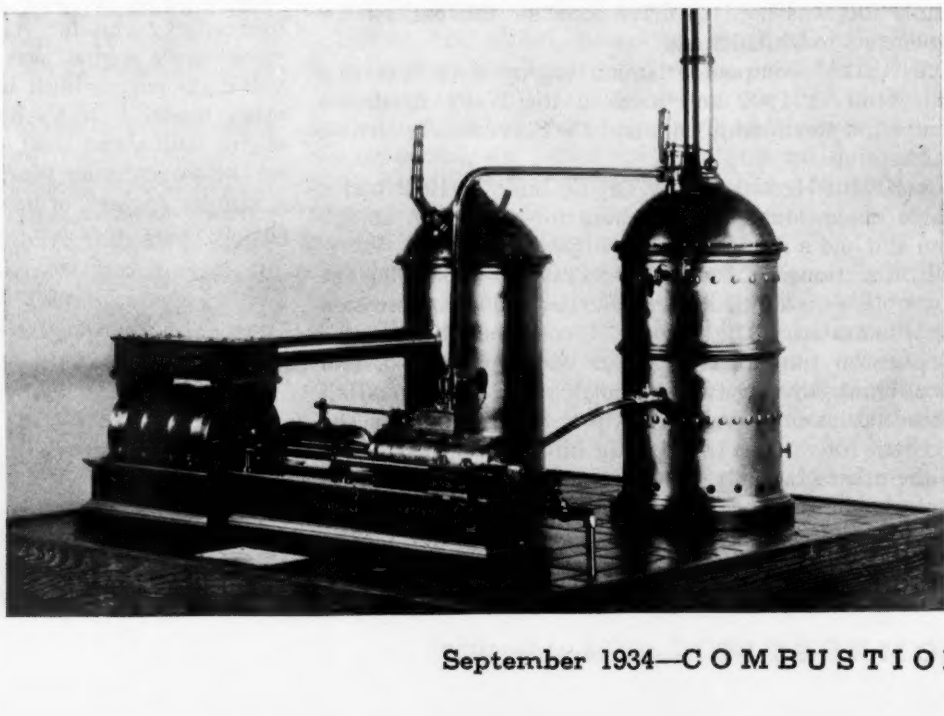
A model of an overshot water wheel operating a power hammer illustrates a type of prime mover suitable for greater water heads, ranging from 10 to 70 ft and in quantities from 2 to 30 cu ft per min. Water enters along a sluice at the top of the wheel and falls in a parabolic path into buckets on a rotor.

The Pelton water wheel was used to drive generators at high speeds. It is a reaction water wheel using a high-pressure stream of water directed against buckets set radially on the periphery of a disk free to rotate. The energy used by the buckets causes the wheel to rotate and develop power.

Muscle power is shown by a dog treadle used in 1881 for churning. Sheep were also used in various parts of the country.

An exhibit of a vacuum vapor power plant built by Cochrane Corporation[1] is a complete self-contained power

[1] Exhibited at the 1932 New York Power Show.



Model of Ross pumping engine which was designed about 1856

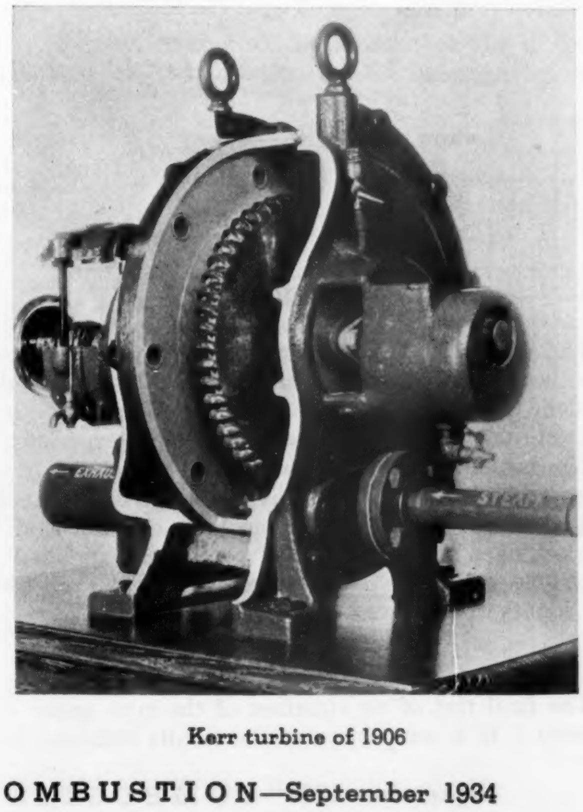
generating plant, consisting of a boiler, a nozzle, turbine wheel and a condenser. Boiler feed is by gravity.

The plant is constructed of two glass coils and a glass boiler and turbine casing. It is operated by the difference in temperature between the coils. One coil is wrapped with gauze and wetted with water from a wick and is thus cooled by the evaporation of the water from its wrappings. This causes a flow of vapor at room temperature within the power plant, which revolves the turbine wheel. A power plant of this type is not practical for commercial purposes, as was demonstrated by the Claude installation in Cuba, but it illustrates that power may be derived from extremely small temperature differences.

A completely detailed model of the Comal power plant at New Braunfels, Texas, on the San Antonio River, built by the U.G.I. Contracting Co., illustrates the successful operation of a power plant using pulverized lignite for fuel. To overcome the difficulties in drying lignite, it is crushed into small pieces on the entrance to a bunker. Below the bunker it is pulverized and then passed through a special air drying process and finally stored in another bunker until ready for use.

There is also shown the John W. Kelly "Hydro Vacuo" engine, built on the supposition that a vacuum in conjunction with water pressure produces more power than is inherent in the water itself. It is not the sympathetic vibration perpetual motion Kelly Motor which was discovered to be a famous fraud. However, sections of the copper tubing used on the fraudulent machine, together with a picture of the machine, are displayed.

The model of the Charles Readhefers perpetual motion machine built by Isaiah Lukens in 1812 was built to deceive and create the impression that power was obtainable by the mere exertion of forces. It is one attempt at the impossible solution of perpetual motion. The original machine was manually operated through a hidden mechanism. The Pennsylvania legislature in 1812 ordered an investigation and proved it a fraud.



Kerr turbine of 1906

Another exhibit, an electrically lighted and operated scenic view made by the Scene In Action Corp. is the Rocky River Hydro-Electric Development. The development built by the United Engineers and Constructors for the Connecticut Light and Power Co. on the Housatonic River in Connecticut is the first American plant to fill a storage reservoir by pumping water at off peak loads. Two 112,500 gpm pumps operated by 8100 hp motors, pump the water from the river, which is 230 ft below to the Rocky River reservoir. The reservoir is 10 miles long and the canal leading from it to the Penstock is $^1/_2$ mile long and 45 ft deep. The reservoir is of insignificant drainage area, and with only natural resources would require five years to fill it. Thus by pumping water with off peak power the 33,000-hp turbine, and 25,000-kw generator, can operate during the period of greatest demand, which is its most efficient operating point.

A miniature model of the Stevenson Dam and power house built by the U.G.I. Contracting Co. on the Housatonic river in Connecticut is complete in all details.

In the liquid flow exhibit are shown seven types of measuring devices, from the simple house meter to the more elaborate orifice method. The setup is such that the entire system can be operated at variable capacities by the visitors who can in this way get an idea of the functions of the meters, indicators and recording devices.

Another exhibit illustrates pressure drops across welded 90 degree ells and screwed ells. Manometers are connected in parallel to illustrate this point. The pressure drop across the smooth joint or welded ell is lower than across a screwed fitting.

The principles of a reed vibration tachometer, and a flyball speed counter are illustrated in action together with a sectioned Lonergan safety valve to show its mechanism in detail.

Other interesting exhibits which have recently been acquired are now being prepared for display.

The author wishes to make acknowledgment to Henry S. Harris, Assistant Associate Director of The Franklin Institute for much information and assistance in securing the photographs.

J. J. Richards has been appointed Manager of Link-Belt Company's Vibrating Screen Department, succeeding Mr. Harry L. Strube, who has been promoted to the position of Assistant Chief Engineer of the company's Philadelphia plants.

Andrew D. Hunt, has been appointed Manager of Engineering of the South Philadelphia works of the Westinghouse Electric & Manufacturing Company, resuming the position which he held from 1926 to 1931 at which time he was temporarily transferred to the Chicago Office of the Company to act as steam specialist for the Northwestern District. In October 1932 he was recalled to the South Philadelphia works to serve again in the capacity of Steam Service Manager and recently resumed his duties as Manager of Engineering.

A Comparison of Psychrometric Equations

By C. HAROLD BERRY

Professor of Mechanical Engineering, The Harvard Engineering School, Cambridge, Mass.

This paper presents a comparison of two well-known psychrometric equations with a modified form of a simpler equation proposed long ago. The comparison indicates that, with ordinary wet-bulb thermometry, the simpler equation is entirely adequate in the range of ordinary atmospheric conditions met in the power plant. The use of the simpler equation is suggested as a means of simplifying arithmetic. This is the second of a series of articles relative to humidity computations, the first of which appeared in the August issue; the third will appear in October.

ABOUT a century ago, James Apjohn[1] proposed an equation for computing the partial pressure of water vapor in the atmosphere from readings of wet- and dry-bulb thermometers. This was,

$$p_v = p_w - \frac{B}{30}\frac{(t_d - t_w)}{87} \tag{1}$$

where p_v is the actual vapor pressure, p_w is the pressure of saturated vapor at the wet-bulb temperature, B is the barometric pressure and t_d and t_w are the dry- and wet-bulb temperatures, respectively, Fahrenheit. The pressure units may be any whatever, so long as all three pressures are in the same units.

Apjohn expressed some uncertainty concerning the value of the divisor 87, and indicated that in some of his experiments he found a value of 89. He attributed this to experimental error, but did suggest that perhaps 88 might be a better value.

In 1886, Professor William Ferrel[2] proposed another equation of the same general form, but with a correction term. This was,

$$p_v = p_w - 0.000367\,B\,(t_d - t_w)\left[1 + \frac{t_w - 32}{1571}\right] \tag{2}$$

This can readily be reduced to a slightly simpler form

$$p_v = p_w - 0.000360\,B\,(t_d - t_w)\left[1 + \frac{t_w}{1539}\right] \tag{3}$$

In 1911, Carrier[3] proposed a somewhat more complicated equation, in the form

$$p_v = p_w - \frac{(B - p_w)(t_d - t_w)}{2800 - 1.3\,t_w} \tag{4}$$

The object of the present paper is to compare these equations. In a previous article[4] the author presented a brief report of an extended study of the accuracy of humidity results, and showed that very small errors in the temperature measurements will produce relatively large errors in the computed results. In view of this, it may be that, with ordinary temperature measurements, which may be in error by several tenths of a degree, the attainable accuracy of result is so low that a simple equation like Apjohn's will serve as well as the more complex equations. If so, it would appear to be proper to take advantage of the simplicity and thereby lighten the tedium of computation.

An Algebraic Comparison

As a first step in comparing these formulations, let it be noted that all three may be reduced to the same form, with the following results:

Apjohn: $$p_v = p_w - \frac{B}{30}\frac{(t_d - t_w)}{87} \tag{5}$$

Ferrel: $$p_v = p_w - \frac{B}{30}\frac{(t_d - t_w)}{N_f} \tag{6}$$

where $$N_f = \frac{92.72}{1 + \frac{t_w}{1539}} \tag{7}$$

Carrier: $$p_v = p_w - \frac{B}{30}\frac{(t_d - t_w)}{N_c} \tag{8}$$

where $$N_c = 93.33\left(1 - \frac{t_w}{2154}\right)\left(\frac{B}{B - p_w}\right) \tag{9}$$

Fig. 1 displays the variation in value of the divisor that has been symbolized by the letter N. It becomes evident that Apjohn's misgivings concerning his value of 87 were not based, as he supposed, upon experimental error, but upon a real departure of actual conditions from those he assumed. It is not surprising that his data seemed to give values as high as 89. Had he worked over a wider range of conditions, and had he been able to measure temperatures with greater precision, his doubts would have been allayed.

A Numerical Comparison

The final test of an equation of the type under discussion is in a comparison of the results obtained from

[1] Transactions of the Royal Irish Academy, Vol. 17 (1837), page 275.
[2] Annual Report of the Chief Signal Officer, U. S. Army, Appendix 24, page 249. The equation as originally presented is not exactly in the form of equation (2), but its equivalent in Centigrade temperatures is in the original Report.
[3] Transactions, A.S.M.E., Vol. 33 (1911), page 1005.
[4] COMBUSTION, Vol. 6, No. 2 (August 1934), page 15.

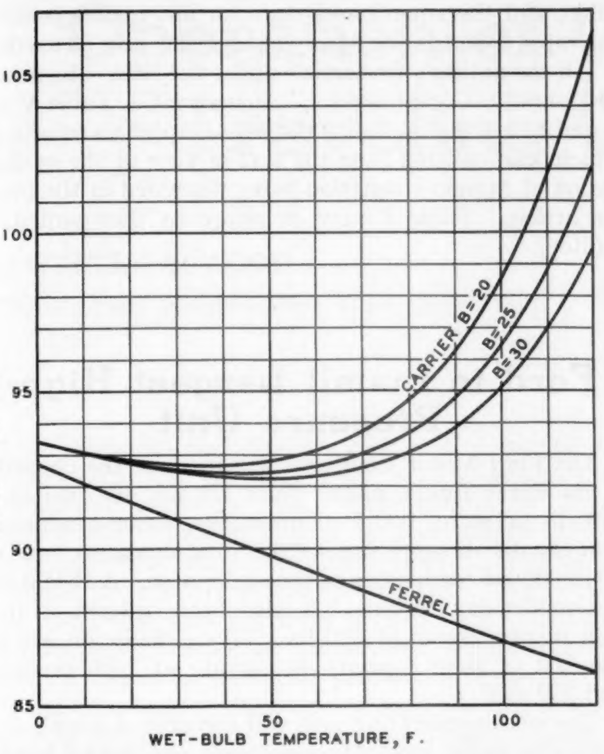


Fig. 1—Values of the divisor of the wet-bulb depression in equation (6), of Ferrel, and equation (8), of Carrier

Values for the Carrier divisor are given for three values of barometric pressure, 30, 25 and 20 inches of mercury at 32 F

identical data. Accordingly, values of t_d, t_w and B have been selected covering the usual range of atmospheric conditions,[5] and for these assumed values the vapor pressure has been computed by the three equations, (1), (3) and (4).

It is immediately evident that Apjohn's equation (1) departs widely from that of Ferrel, and still more widely from that of Carrier. Carrier gives values higher than Ferrel, and Apjohn gives values decidedly lower than Ferrel.

The algebraic comparison indicated that the divisor 87 proposed by Apjohn is too small. From an inspection of Fig. 1, it would appear that perhaps 90 might give more satisfactory results. Accordingly, in this numerical computation there has been included Apjohn's equation, modified to the divisor 90.

$$p_v = p_w - \frac{B}{30}\ \frac{(t_d - t_w)}{90} \qquad (10)$$

The numerical values are presented in Table I, and Fig. 2 gives a plot of differences between Carrier and Ferrel, respectively, and Apjohn with the divisor 90. This plot also includes curves indicating the departure in vapor pressure that would result from an error of 0.1, 0.2 and 0.3 deg. F in the reading of the wet-bulb temperature. The results of Apjohn's original equation (1) are not included, since they depart widely and do not appear significant.

Examination of the plot of Fig. 2 indicates that the greatest departure of Carrier from Apjohn with divisor 90 is less than the departure caused by an error of 0.3 deg F in the wet-bulb temperature. The greatest departure of Ferrel is about equivalent to an error of 0.14 deg F in wet-bulb temperature.

Conclusion

The conclusion seems to be obvious that, in the range of ordinary atmospheric conditions, if the wet-bulb temperature is measured with a probable error of $^1/_4$ deg F or more, there is no justification for using the more complex equations of Ferrel or Carrier, but that equally dependable results will be found from Apjohn's simpler equation, modified to the divisor 90.

It is further evident from the error curves in Fig. 2 that the variations due to small errors in wet-bulb temperature will amount to several thousandths of an inch in the vapor pressure. Accordingly, vapor pressure should not be computed beyond the second decimal place (hundredths of an inch of mercury) unless the wet-bulb temperature has been measured with a probable error of considerably less than 0.1 deg F. The

[5] It is recognized that the occurrence of wet-bulb temperatures as high as 100 F is unusual, to say the least. However, the comparison, as a matter of pure arithmetic, has been carried to this value. The results have been computed to four significant digits to bring out the differences, on the assumed data, which are taken to be exact.

Fig. 2—Differences between the Carrier and Ferrel equations and modified Apjohn equation

Carrier values are higher than Apjohn. The curves are marked with the letter C.

Ferrel values are lower than Apjohn, except at low temperatures. The curves are marked with letter F.

The numbers following the letters marking the curves indicate the dry-bulb temperature.

The dotted curves indicate the change in vapor pressure due to an error in the wet-bulb temperature reading, of 0.1, 0.2, 0.3 deg F, as marked.

Barometer = 30 inches of mercury at 32 F. Curves drawn for B = 25 and 20 show substantially the same relations.

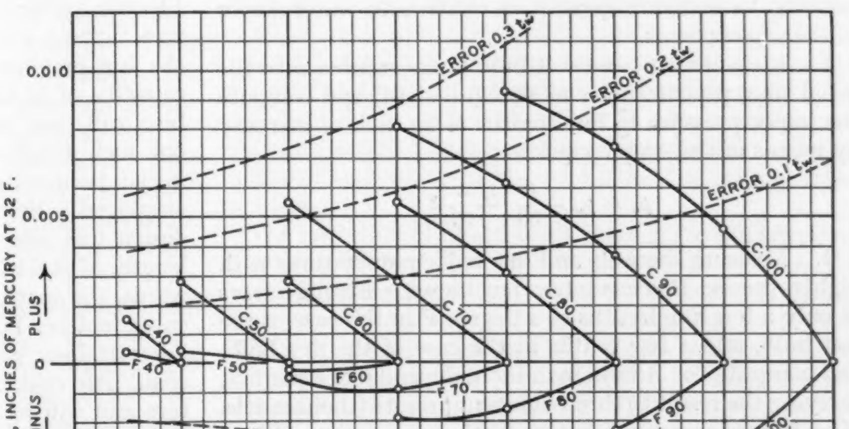


TABLE I—VALUES OF VAPOR PRESSURES COMPUTED BY THE THREE EQUATIONS:

CARRIER, EQUATION (4);
APJOHN (DIVISOR 90), EQUATION (10);
FERREL, EQUATION (3)

Dry-bulb temp. F	Wet-bulb temp. F	Barometer, inches of mercury at 32 F 30	25	20
100	100	1.9314 throughout		
	90	1.3145 1.3099 1.3068	1.3331 1.3284 1.3258	1.3518 1.3469 1.3449
	80	0.8168 0.8094 0.8046	0.8538 0.8464 0.8424	0.8909 0.8835 0.8803
	70	0.4146 0.4053 0.4004	0.4699 0.4608 0.4567	0.5253 0.5164 0.5131
90	90	1.4210 throughout		
	80	0.9242 0.9205 0.9182	0.9427 0.9390 0.9370	0.9612 0.9575 0.9560
	70	0.5226 0.5164 0.5132	0.5595 0.5534 0.5507	0.5964 0.5905 0.5883
	60	0.1962 0.1881 0.1853	0.2516 0.2436 0.2412	0.3067 0.2992 0.2973
80	80	1.0316 throughout		
	70	0.6306 0.6275 0.6259	0.6490 0.6460 0.6446	0.6675 0.6645 0.6634
	60	0.3047 0.2992 0.2973	0.3416 0.3362 0.3347	0.3782 0.3733 0.3720
70	70	0.7386 throughout		
	60	0.4131 0.4103 0.4094	0.4315 0.4288 0.4280	0.4498 0.4473 0.4467
	50	0.1457 0.1402 0.1397	0.1822 0.1772 0.1768	0.2188 0.2143 0.2139
60	60	0.5214 throughout		
	50	0.2541 0.2513 0.2511	0.2723 0.2698 0.2696	0.2906 0.2883 0.2882
50	50	0.3624 throughout		
	40	0.1395 0.1367 0.1371	0.1577 0.1552 0.1556	0.1759 0.1737 0.1740
40	40	0.2478 throughout		
	35	0.1495 0.1480 0.1484	0.1586 0.1573 0.1576	0.1677 0.1666 0.1668

permissible errors in dry-bulb temperature range from two to ten times those in the wet bulb.

Thus there are two courses open to the engineer desiring to measure quantities relative to water vapor in the atmosphere:

1. Measure dry- and wet-bulb temperatures with the usual sling psychrometer or its equivalent, and compute the vapor pressure to hundredths of an inch of mercury by means of the simple equation

$$p_v = p_w - \frac{B}{30} \cdot \frac{t_d - t_w}{90}$$

2. Measure the wet- and dry-bulb temperatures with highly precise thermometric technique, ensuring errors of only a few hundredths of a degree F in the case of the wet bulb, and a few tenths in the case of the dry bulb, and compute the results by a more dependable equation, carrying the result to three significant digits (thousandths of an inch of mercury), but no further.

It is believed that the simple modified Apjohn equation is adequate for almost all engineering work involving atmospheric humidity, and it is further believed that its simplicity is such that it is easier for the occasional worker to use it than to hunt up one of the established humidity charts. This simple equation, together with the steam table, and the equation of state for air, makes possible the rapid computation of any results that may be needed.

[In the author's first article under the title, "Accuracy of Humidity Computations," August, 1934, Table V referred to matter discussing the use of Apjohn's equation, which was omitted from the text in view of the modification of Apjohn's equation being discussed in the present article. Table V may therefore be disregarded.—Editor].

Ford to Install Largest High-Pressure Unit

The Ford Motor Company will increase the capacity of its River Rouge power plant by the addition of a 1400-lb pressure, 900 F temperature, steam-generating unit capable of supplying 800,000 lb of steam per hour to a 110,000-kw condensing turbine-generator. A 15,000-kw non-condensing turbine has also been ordered to furnish process steam at 250 lb. This installation will be the first of large capacity to operate at high pressure and 900 deg.

The steam-generating unit will comprise a double-set, bent-tube boiler, having forged drums with welded heads, water-cooled furnace, superheaters, air heaters, economizers and pulverized fuel burning equipment, to be furnished by Combustion Engineering Company, Inc. It will replace one of the older, low-pressure boilers and produce nearly three times the quantity of steam in the same floor space at improved efficiency. In general design it will be similar to the two high-pressure units installed in 1929, each rated at 700,000 lb of 750 deg steam, except that it will be somewhat larger and operate at higher steam temperature. The furnaces will be completely water-cooled, corner-fired and have double slag screens. Pulverized coal will be supplied by the existing coal preparation plant and storage system.

Very liberal fan capacity has been provided so that steam outputs in excess of that stated should be possible if desired.

The 110,000-kw turbine-generator will be a General Electric vertical-compound machine with the high-pressure turbine and generator mounted directly on top of the low-pressure element. Each element will have a capacity of 55,000 kw and run at 1500 rpm. The turbine will take steam at 1200 lb, 900 F and exhaust at one inch absolute back pressure. In view of the high initial temperature no reheating of the steam will be employed. Because of limited floor space the vertical design was selected and the overall dimensions will be length, 57 ft 6 in., width 23 ft and height 21 ft. In other words, the space occupied will be less than a quarter of a cubic foot per kilowatt of output. It is expected that a kilowatt hour will be produced on less than a pound of coal. Air coolers will be built integral with the generators and will be arranged so that the heat losses in the generators will be recovered in the boiler feedwater. The new turbine-generator will be similar in many respects to the 110,000-kw unit installed in this plant in the early part of 1931.

It is understood that the additional power made available by the new installation will be utilized in expanding the steel mill facilities at the River Rouge Works.

Operation of Chain Grate and Traveling Grate Stokers*

By WALTER H. WOOD

Combustion Engineering Company, Inc.

When stokers are installed the operator is usually provided with such instructions as will produce the best operating results. But personnel often changes, instructions are lost or filed away, and the installation may not be operated at best efficiency. The present article gives practical information of a basic character that will assist operators of chain grate or travelling grate stokers.

IT IS estimated that the cost of the fuel burned on chain grate and traveling grate stokers annually in industrial plants in the United States is about $25,000,000. Through lack of skill or proper care in operation, there is a preventable loss of this fuel which amounts to probably $2,000,000 per year. While even that large sum may not seem so important when the number of plants involved is considered, the fact remains that the losses in individual plants may easily run to several thousands of dollars per year. The fuel is a raw material put in the hands of men to be used in making steam, and often there are inadequate means to determine whether this material is wasted or used properly. It does not go into a finished product such as cloth or paper or food which can be examined or tested. The men who operate the stokers which burn the fuel are usually the only ones who are in a position to know whether much or little of the fuel is being wasted, and how and why the wastes occur. There is no system of combustion regulation that can regulate all fuel wastes.

The management of an industry often comprises men who have had the experience in the manufacturing and marketing of a product necessary to keep the industry on a paying basis, but who know little or nothing about steam production except that there are large annual costs of fuel, labor and maintenance and that production is sometimes slowed down because of power plant difficulties. Too often the management leaves the operation of the vitally important steam generating plant to the judgment of men who are none too well qualified to discharge properly the duties intrusted to them. This applies particularly to the matter of fuel burning. In order to keep the cost of labor in plant operation down, low priced boiler room help is often employed and in many instances such low priced help is expensive. Unless the operators are skilled and interested in their work much of the raw material which is used by these operators will be wasted without any one knowing the amount of or reason for the waste.

As long as a plant is in operation fuel must be burned and as long as fuel is burned there is danger of unnecessary loss of fuel through poor operation. Hence the importance of continual watchfulness on the part of the operators if maximum plant economy is to be maintained. Responsibility for the failure to maintain good economy does not always rest with the men who operate the stoker equipment. Often such men are left to their own judgment and resources without having been given proper guidance in the art of fuel burning. Manufacturers of fuel burning equipment usually have men available who are well qualified to instruct plant operators in this art, and the advice of such men should be had from time to time, especially if changes are made in operating personnel, as such advice will usually result in appreciable fuel savings.

There are three principal losses which a skilled operator will try to keep at a minimum. These are (a) loss of carbon rejected with the ash into the ashpit or carried with the gas stream into the boiler setting, (b) loss due to excess air supplied to burn the fuel (low CO_2) and (c) loss due to unburned combustible gases leaving the furnace. An attempt to reduce any one of these losses too much will result in an increase in one or both of the other two, and it is therefore necessary to balance the three in such a way that their sum will be a minimum. If too high CO_2 is carried in order to reduce or counteract the excess air, a loss may occur due to unburned combustible gases (CO) and possibly to increased carbon in the ashpit refuse, while low ashpit refuse loss may be obtained by using an excessive amount of air in burning out the carbon in the fuel bed.

The operation of forced-draft chain grate and traveling grate stokers should be considered first with regard to the kind of fuel used and second as to the type of furnace arrangement employed. In general fuels can be divided into two classes, free burning bituminous coals comprising one class and anthracite buckwheat and coke breeze the other. The furnaces are also divided into two classes, usually referred to as front arch and rear arch types. There is a third arrangement in somewhat general use comprising both a front and a rear arch. Such arrangements are usually classed as rear arch furnaces.

Bituminous coals suitable for use on chain grate and traveling grate stokers are usually burned in front arch furnaces although some of the more recent installations

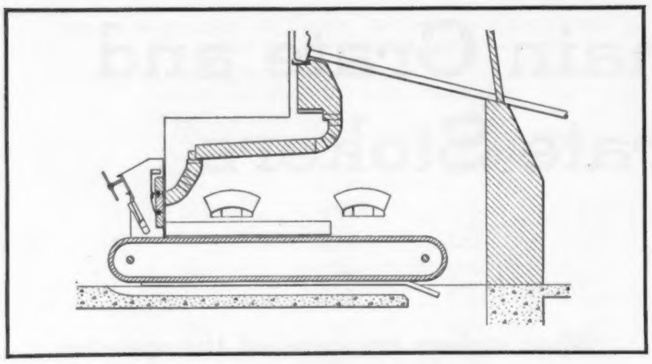


Fig. 1—Typical front arch furnace for free-burning bituminous coal

are of the front and rear arch type. Fig. 1 illustrates a typical front arch furnace for use with free-burning bituminous coals, while Fig. 2 shows a furnace with both front and rear arches, these arches being of the water-cooled type. The bituminous coals most generally used on chain and traveling grate stokers are those which are mined west of the Appalachian mountains, commonly referred to as the mid-west and western bituminous coals. Like all bituminous coals, these fuels coke to a greater or lesser degree although they are referred to as free-burning coals. When these coals are burned with their normal low moisture content, the ashpit refuse contains enough carbon to constitute a serious loss and for that reason it is advisable to add moisture to the fuel to bring the total moisture content up to about 10 per cent to 15 per cent. Best results are obtained when the moisture is added before the fuel reaches the stoker hopper, in order to have as uniform mixing of the fuel and added moisture as possible. Fire thicknesses should range approximately from 4 to 7 in., best results being obtained when the fuel bed is about $5^1/_2$ to 6 in. in thickness.

The air pressures required for good combustion depend on the sizing of the fuel, speed of ignition, furnace design and fuel bed thickness. In general, it may be said that a 6-in. fuel bed of 1-in. nut and slack having 10 to 12 per cent moisture, will require about 0.7 to 1.0 in. in the first compartment. Pressures in succeeding compartments should be somewhat higher until the fuel bed gets thin when the air pressures should be lowered. After ignition is established over the first stoker compartment, the fuel should be burned as rapidly as possible, care being taken to prevent lifting fuel off the bed by the use of too high air pressures. Small islands or patches of coke at the rear of the furnace cannot be burned out completely without the use of entirely too much air for good results.

In burning bituminous coal with front arches a large amount of combustible gas is distilled off at the front of the furnace which probably has a reducing action on the metallic oxides in the refractories, as more or less serious erosion frequently occurs. Secondary air over the fuel bed, introduced through the arch, aids in reducing the erosive action and at the same time produces turbulence in the gas stream which increases the efficiency of combustion and reduces the amount of objectionable smoke.

There are some bituminous coals mined in western Pennsylvania and in West Virginia which are very suitable for use on forced draft chain grate and traveling grate stokers. These are not quite as free burning as western and mid-west bituminous coals, and the addition of moisture does not reduce carbon losses in the ashpit to the same extent. Ashpit losses resulting from burning the western Pennsylvania and West Virginia coals are lower if the coal is sized to pass through a $^3/_4$-in. screen with the fines below about $^1/_4$ in. removed.

In general, the following suggestions apply to the burning of bituminous coals on forced-draft chain and traveling grate stokers:

1. The fuel should be crushed to pass through a $1^1/_2$-in. screen as a maximum and $^1/_2$ to $^3/_4$-in. as a minimum, and in the case of mid-west and western bituminous coals, the total moisture content should be about 12 to 15 per cent. If the fuel is western Pennsylvania or West Virginia free burning coal, little or no advantage will be found in adding moisture to reduce the amount of coke in the ashpit. If that loss is excessive it can usually be reduced by screening out the coal under $^1/_4$ in. in size.

2. The coal should be burned as rapidly as possible after ignition is well established, care being taken to prevent the loss of carbon by the use of too high air pressure in any compartment. The maximum air pressure should be applied in the second and usually about the same pressure in the third air compartments, reducing these pressures toward the rear of the stoker as the fuel bed burns thin. Very little air is generally needed in the extreme rear compartment, and too much air is responsible for high excess air loss (low CO_2). The CO_2 at the boiler damper, if the setting is reasonably tight should not be under 12 per cent for best results, while CO_2 above 14 per cent at the boiler damper is likely to cause excessive furnace temperatures, rapid failure of refractories, and some loss due to unburned combustible gases.

3. Thickness of the fuel bed depends upon sizing of the coal. Good results at combustion rates up to 40 or 45 lb of coal per sq ft of grate per hour can be obtained with fuel beds $5^1/_2$ to 6 in. in thickness.

4. Unless the load on the stoker varies widely it is not advisable to change the thickness of the fuel bed, once a thickness which gives good results has been established. The variations in load demand over a rather wide range can be taken care of by varying the length of the fuel bed in a front arch furnace.

5. Air pressure gages should be connected to each

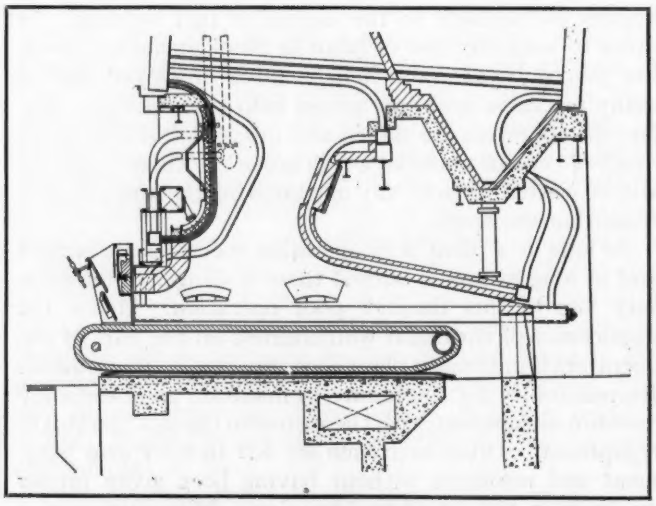


Fig. 2—Furnace with both front and rear water-cooled arches

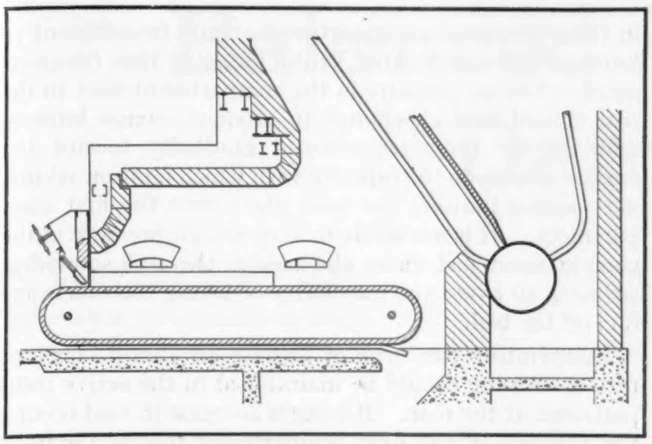


Fig. 3—Front arch furnace for burning buckwheat or coke breeze

stoker compartment, and so located that the operator can observe the changes in air pressure at the time he operates the dampers.

6. The rear of the stoker should be watched frequently to make sure that the fuel does not go into the ashpit unburned.

7. The furnace draft should not be low enough to produce a pressure in the furnace, nor great enough to reduce the CO_2 in the gases, due to leakage of air into the furnace.

The burning of anthracite buckwheat and coke breeze requires more careful attention than is needed in burning bituminous coals if the best results are to be had. Except for the fact that a coke breeze fuel bed should be somewhat thicker than an anthracite fire, the two fuels should be treated very much alike. These two types of fuels are burned in both front arch and rear arch furnaces. The methods employed in operating the two types of furnaces are decidedly different and should be studied separately.

Fig. 3 shows a typical front arch furnace for either of these fuels, while Fig. 4 illustrates a combined front and rear arch furnace. Fig. 5 is typical of the most satisfactory rear arch furnaces for these fuels.

No. 3 buckwheat is the size of anthracite buckwheat most commonly burned in front arch furnaces. It is prepared to pass through a $^3/_{16}$-in. round mesh screen and lie on a $^3/_{32}$-in. round mesh screen. There is always more or less undersize in this fuel. For best results with front arch furnaces this undersize should not exceed 20 per cent, as the finer sizes are easily lifted from the fuel bed with the air required for burning, and lost in the ashpit or in the boiler setting. The most satisfactory fuel bed thickness for No. 3 buckwheat is usually 3 to $3^1/_2$ in. If the fuel bed is too thin solid carbon and excess air losses are too high and if the fuel bed is too thick the loss due to unburned combustible gases is excessive. It is difficult to maintain an even fuel bed if it is too thick, as holes will develop in the fire and result in a combination of the three major losses referred to above.

The pressure of air supplied for burning the fuel should be sufficient to lift fine particles an inch or two above the bed and produce a "shimmering" appearance when observed with a colored glass or by closing the eyes until the fire can be seen through the eye lashes. Care must be used to keep the air pressure in the first compartment just low enough to avoid disturbing ignition. If small black spots appear on the surface of the fuel bed, or if small black particles can be seen blown up from the bed the air pressure is too high and should be reduced. Air pressures in the second and third compartments should be somewhat higher than in the first compartment unless a very short fire is carried in which case the pressure in the third compartment should be lowered. Pressures should be reduced toward the rear of the stoker sufficiently to avoid causing a shower of fuel to be lifted from the bed and blown to the rear of the stoker or into the ashpit. The air pressure should never be low enough in any compartment to produce a flat incandescent surface with a plastic or waxy appearance, as such a condition results in a high loss due to the distillation of combustible gases which will not be burned completely in the furnace. Frequently such fuel beds are maintained, and rather violent burning is done on the rear one or two compartments. Such a method of firing is very commonly found and is always an indication of very poor operating results. The entire fuel bed must be kept active without blowing an excessive amount of fuel into the ashpit or boiler setting if best results are to be expected.

In order to maintain proper air pressure conditions in all compartments, a constant pressure should be maintained in the air duct, and pressure changes in individual compartments should be made by opening and closing the compartment dampers just enough to produce the required fuel bed condition. Wide changes in compartment pressures, say over 0.2 in., should be avoided as these give rise to solid carbon loss or to the production of unburned combustible gases. Where automatic regulators are installed in connection with stokers burning anthracite buckwheat, the regulators should be adjusted to produce gradual and not wide changes in air pressure.

The foregoing suggestions apply to the burning of coke breeze as well as to buckwheat. The breeze should be about $^1/_2$ in. in size and contain at least 20 per cent undersize, say $^1/_8$ in. and smaller. If the breeze is coarse and free of undersizes, the fuel bed will be too porous and difficulty will be experienced in maintaining ignition.

The combined front and rear arch furnace give very good efficiencies with No. 3 buckwheat and coke breeze especially at combustion rates up to 25 to 28 lb per sq ft of grate per hour. If the combustion rate exceeds about

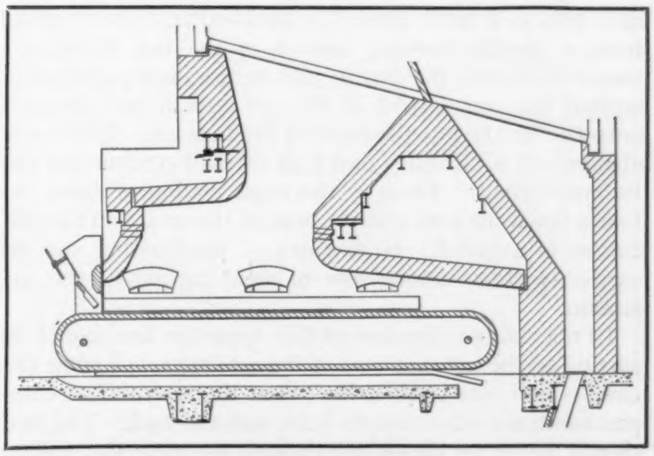


Fig. 4—Combined front and rear arch furnace

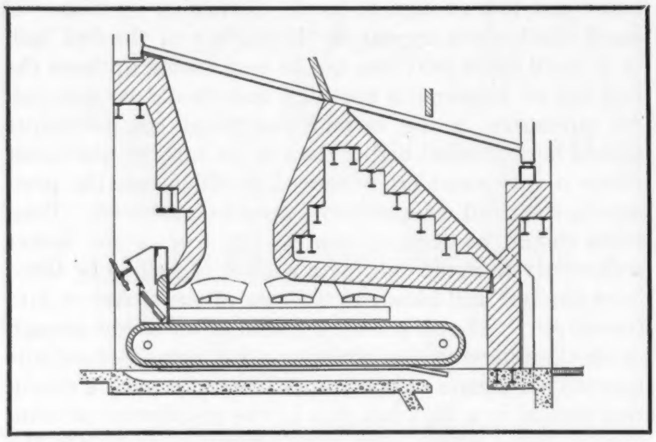


Fig. 5—Most satisfactory rear arch furnace for buckwheat or coke breeze

30 lb it becomes necessary to burn much of the fuel on that part of the stoker immediately under the throat between the arches and small sized particles of fuel are lifted out of the bed and carried unburned into the boiler setting. In operating such a furnace the air pressures in the two rear compartments can be high enough to burn the major part of the fuel at a high rate, although some air should be used in the forward and middle compartments. Ignition is more stable in this type of furnace than in a front arch type and therefore considerable burning can be done at the forward part of the furnace.

The use of a relatively large amount of air at the rear of the stoker prevents high ashpit losses. The gases from that part of the furnace contain excess air while those from the forward part of the fuel bed are rich in combustible matter, and these gases are well mixed in the throat between the arches. In that way CO and excess air losses are low, and stratification of gases is avoided. Many operators make the mistake of trying to burn too much fuel on the rear and too little at the front of this type of furnace. As in the case of the front arch furnace the fuel bed must be kept active over the entire area, with most of the burning however, at the rear of the fuel bed.

The single, rear arch furnace shown in Fig. 5 is probably the simplest of all from the standpoint of operation, once the correct method of operation is understood. The results obtained from such installations are better over a wide range of combustion rates than with any other type of furnace. When properly constructed, the arch acts as a baffle above the fuel bed and directs gases from a rapidly burning portion of the bed at the extreme rear over the entire fuel bed. Fine particles of ignited fuel are carried in the gas stream and dropped onto the fuel bed at the front of the furnace. Ignition is always well established and high rates of combustion can be maintained. There is thorough mixing of gases between the front wall and the nose of the arch and stratification is avoided. High rates of combustion can be carried without undue loss of solid carbon in the gas stream.

In operating a furnace of this type the fire should be carried to the extreme rear of the stoker except when the combustion rate is quite low; then the next to last compartment may be used to burn out the fuel. The bed should never be shortened enough to bring the end of the fire up near the nose of the arch. The air pressure in the extreme rear compartment should be sufficient to burn out the combustible in the fuel over that compartment. The air pressure in the compartment next to the rear should be high enough to produce intense burning and the air pressures reduced gradually toward the front. Owing to the rapidity with which ignition occurs, very active burning can take place over the first compartment. It is advisable to keep the air pressure in the compartment just under the nose of the arch somewhat reduced, to avoid the possibility of lifting too much fine fuel off the bed.

In operating this type of furnace an almost constant rate of burning should be maintained in the active compartment at the rear. If a slight increase in load occurs, the pressure in the next compartment toward the front should be increased until it is nearly equal to that in the rear active compartment. Further increases in load should be taken care of by increasing the pressure in the successive compartments toward the front. Decreases in load should be made by reversing the operation, reducing the pressure first in the forward compartments, and working back toward the rear of the stoker, but always maintaining some air pressure in all of the compartments.

A fuel bed for good results in such a furnace can be 4 to 5 in. thick with No. 3 buckwheat. If the fuel bed appears to increase in thickness about the middle and toward the rear, with a decided increase in thickness next to the active compartment at the rear, there is not a proper distribution of air in the compartments, too much air being used in the rear one and not enough in those toward the front. Such a fuel bed if observed through a colored glass will be found to build up at the rear and avalanche onto the rear compartment. Too much fuel is being blown off the grate, and that fuel cannot be carried in the gas stream but drops out early causing the thickening of the bed at the rear. The condition can be corrected by reducing the air pressure in the last active compartment and increasing it in the compartments toward the front. After a time it will be seen that the fuel bed begins to thin down and finally shows a gradual reduction in thickness from the mid portion toward the rear of the furnace.

The suggestions which have been made, relating to the operation of stokers are of a very general nature, as specific instructions cannot be given to cover all conditions as to fuels, furnace design, load variations, etc. An effort has been made to set up a guide which an operator can use in working out a method of operation that will be best suited to his particular problem of burning fuel with as little waste as possible. What should be kept in mind is that fuel losses will occur whenever fuel is burned, and that the amount of the losses can be controlled to some extent by the operator. It rests with him in a very large measure as to whether the losses will be great or small. Unless skill is used and close attention is given to the work, there will continue to be inexcusable losses of fuel. If as much care on the part of a stoker operator is exercised in burning fuel as is given the manufacture of some article by a skilled workman and as close supervision shown by the engineer in charge of a boiler room as by a department superintendent in a well organized industrial plant, a surprisingly large amount of fuel will be saved. Eternal vigilance is the price of good economy in any boiler plant.

Balanced Regulating Dampers

By CHARLES B. ARNOLD
Brooklyn Edison Company

A review of the article on balanced regulating valves written by I. E. Church has prompted the author to relate similar experiences on so-called balanced dampers at the Hudson Avenue Station. Studies were made as to the relation between torque required to move a damper and damper position for various pressure drop conditions across the damper.

As Mr. Church of the Duquesne Light Company has very completely described, in the August issue of COMBUSTION, the unbalanced forces which are found in the so-called balanced regulating valves have offered many operating difficulties to the users of these valves. The experience, with some designs of feed valves had by the Brooklyn Edison Company at its Hudson Avenue Station have been much the same as those experienced by Mr. Church. These valves in the nearly closed position would snap closed as a result of the cavitation effect of the water passing through them and then immediately open and thus produce a severe water hammer in the feed line and render unsatisfactory regulation. This trouble was eliminated by a redesign of the ports in a manner similar to one of the design changes cited by Mr. Church.

In much the same manner as flow of water through a regulating valve produces unbalanced forces on the moving parts of the valve, so also does the flow of gas or air through a so-called balanced regulating damper produce similar unbalanced forces. In cases where the draft drop across the damper is high these unbalanced forces sometimes cause very unsatisfactory operation, whether the damper be controlled manually or by some type of regulator. For example, in closing the so-called balanced damper little difficulty is had until a certain point on the travel is reached at which time the unbalanced forces acting upon the damper become sufficient to overcome frictional forces and suddenly move the damper through an appreciable amount by taking up the lost motion in the operating mechanism. This condition produces a sudden decrease in air flow which is very unsatisfactory from a control standpoint. It has been found on some occasions that the unbalanced forces on a damper of this type have been so high that the hydraulic operating cylinder, adequate for normal conditions, was incapable of opening the damper when the pressure drop across it approached the designed maximum available draft. During automatic operation, when a damper which is subject to the above characteristics is caused to close down considerably in response to a change in air flow requirements, it then fails to open again due to lack of available force when the regulator is actuated in the reverse direction.

These operating difficulties led to an investigation into the causes and possible corrections of the trouble experienced. To this end the relation between torque required to move a damper, and damper position was determined for various pressure-drop conditions across the

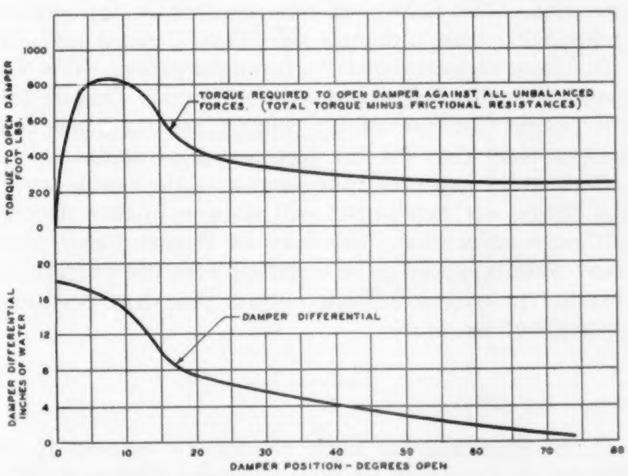


Fig. 1—Characteristics of two-leaf outlet damper having equal areas either side of damper shafts

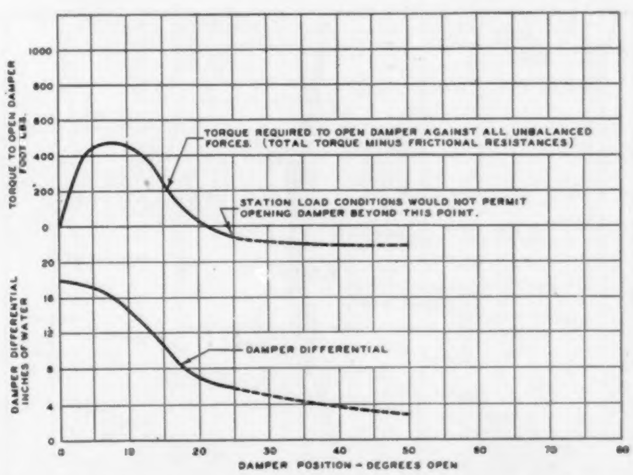


Fig. 2—Characteristics of two-leaf boiler outlet damper having damper areas unbalanced in the ratio of 46:54 in an opening direction

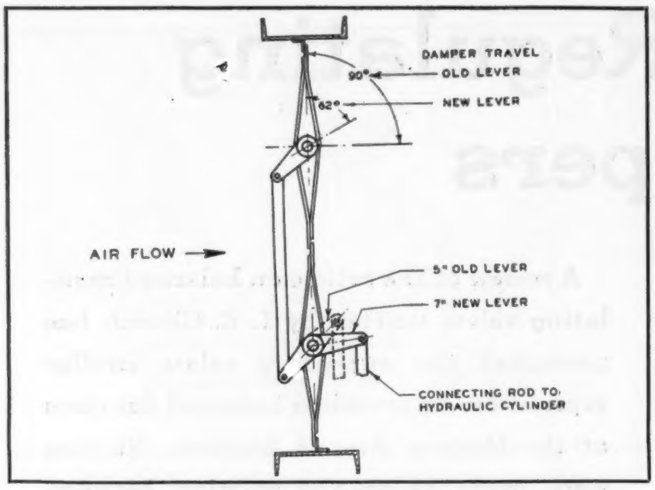


Fig. 3—Method of increasing mechanical advantage of operating levers

damper. The torque required to operate a two-leaf balanced boiler outlet damper with frame dimensions about 24 ft × 4 ft 6 in., is shown in Fig. 1. It will be seen from this curve that the highest torque was required to pass the damper beyond a position 8 deg from full closed. As the opening was further increased, the pressure drop across the damper rapidly decreased thereby decreasing the velocity of the gas flowing by the damper leaves, thus relieving the unbalancing pressures. These facts led to a change in design of subsequent boiler outlet dampers whereby more area was added to the halves of the dampers which turned in the direction of flow than those halves of the dampers which turned against the direction of flow, with the result that some of the unbalanced force was cancelled out. The ratio of these unequal areas was 46:54. Comparing the results of tests as shown in Fig. 2 with those shown in Fig. 1 it will be noted that the maximum torque required to overcome the unbalanced forces of this new design of damper was reduced by about 400 ft-lb. It will be seen also from Fig. 2 that beyond a 20 deg position, a torque was required to restrain its movement in an opening direction. This condition, however, did not offer operating difficulties because the friction of the damper was in excess of this unbalanced force.

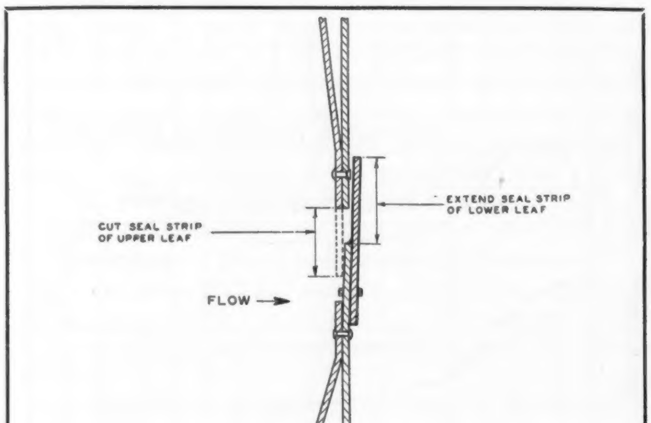


Fig. 4—Method employed to unbalance boiler inlet damper areas in ratio of 48:52 in an opening direction

Less elaborate tests on the boiler inlet dampers of similar two-leaf construction at the entrance to the stoker windbox from the forced-draft duct indicated that extreme torques had to be applied to open these dampers. For one group of dampers which were required to operate under moderate draft differentials, the forces required to open them were brought within the range of the power cylinders by re-arranging the damper operating mechanism as shown in Fig. 3 so that the highest mechanical advantage was had when the dampers were in the nearly closed position. This made it impossible to open the dampers more than 62 deg, but since the remaining 28 deg of damper travel had so little influence on the regulating ability of the dampers, this restriction in damper travel offered no disadvantage.

For another group of two-leaf boiler inlet dampers, the additional force afforded by arranging the linkage, as described above, was insufficient. In order to correct this condition a strip was cut from the lower edge of the upper damper leaf and another strip was welded to the upper edge of the lower damper leaf as shown in Fig. 4. This change rendered a ratio of damper areas either side of the shafts of about 48 : 52 and tests revealed that the maximum forces required to open the dampers were reduced by half.

Caution had to be exercised in attempting to make these changes on the damper as flow conditions, pressure drops and the shapes of the dampers all affected the problem.

It will be noted that the modified design permits slight adjustment of the damper leaf ratio without affecting the main damper construction. In the cases cited the modification was carried far enough to bring the forces within satisfactory operating limits, and no attempt was made to provide a more nearly absolute balance.

Electrical Output Declines

The increase in electrical output for the central station industry over 1933, which has been sustained for a number of successive weeks, failed to hold up for the week ending September 1st. The output was 1,626,881,000 kwhr as compared with 1,637,317,000 for the corresponding period of last year, representing a decline of 0.6 per cent. The falling off was greatest in the central industrial region although the West Central and Pacific Coast regions showed substantial gains. New England was also below last year's figures. Despite this the output for the entire country was 12.7 per cent greater than that for the like period of 1932.

It is to be expected that because of the textile strike the figures for September will show a further decline. Although unfortunate, this may be regarded as a transient condition not to be confused with the progressive increase in central station output that has pertained throughout the year.

L. S. Stephens has been elected to succeed D. B. Piersen as president of the Stephens-Adamson Mfg. Co., conveyor manufacturers of Aurora, Illinois. Mr. Piersen has been active in the firm since 1914.

Reducing Turbine Fire Hazards

To reduce the oil fire hazard from turbine lubrication and governing systems, as became evident through several disastrous fires two or three years ago, extensive changes have been made in the piping of later installations as well as some of the earlier installations. These include the use of welded connections for piping and fittings, remote control of valves and pipe covering impervious to oil on oil lines adjacent to high-temperature steam lines. In one recent installation the pipes carrying oil under pressure are enclosed within the drain pipes and the oil reservoir is located in a completely fire-proof room at the basement level. Non-inflammable fluids are also being employed for governor control and also have some lubricating possibilities.

THE recent Prime Movers Report on "Turbines" contains statements from the Allis-Chalmers Mfg. Company, the Westinghouse Electric & Mfg. Company and the General Electric Company relative to means for reducing fire hazards in the governing and lubrication systems of steam turbines. The first named summarizes the three principal fire hazard problems and discusses their solution in the following statement:

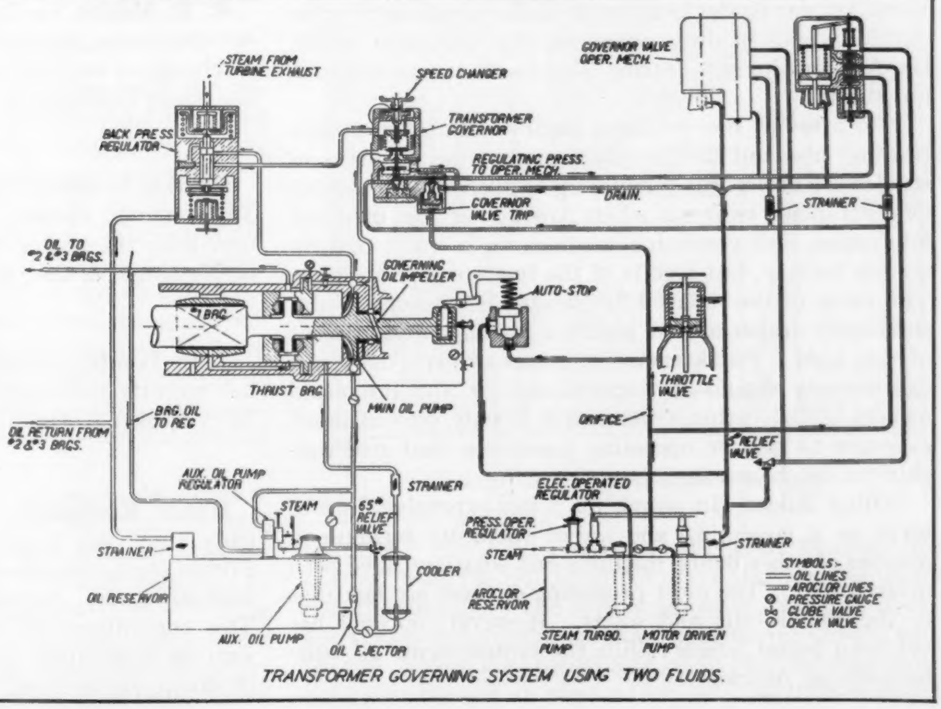
"First—arranging all oil piping and receptacles so that there would be no steam pipe or hot surfaces in proximity to the oiling system and, in addition, whenever required, installing protecting walls between, or sealed covering around, pipes and receptacles.

"Second—locating all oil piping, receptacles and structures on foundations, etc., where little or no vibration occurs.

"Third—use of a large factor of safety throughout the design of the oiling system.

"In the latest design these problems have been solved in the following manner: All oil piping with oil-operated control gear for the steam inlet valves is placed on one side of the turbine; the steam chest, steam inlet valves and steam piping on the opposite side. The oil tank located at the high-pressure end on the high-pressure foundation is arranged so that it can be separated by a wall from the steam pipes and other high-temperature parts. Inlet valve control gear is connected to the inlet valves by mechanical means consisting of a shaft, lever and connecting rods. The piping and receptacles (oil tank, etc.) are supported either on the bedplates or on the foundations, except where pipes are connected to the bearing pedestals or relay valve mounted on the high-pressure pedestal. With this arrangement there will be a minimum of vibration in the whole oiling system. All piping, fittings and valves are made of steel with welded flanges. For oil pressures up to 20 lb per sq in. gage 125-lb flange standard is used with full face gaskets. For pressures above 20 lb per sq in. gage and up to 75 lb per sq in. gage, 250-lb flange standard is used with male and female flanges and ring gaskets. If any further special protection is necessary it can be arranged quite readily as the piping and other parts of the oiling system are separated from the turbine unit itself, except where piping connects to bearing pedestals.

Diagram of Westinghouse governing system employing two fluids, Aroclor for all valve operations and oil for actual governing and bearing lubrication



"Development of non-inflammable fluids suitable for governor control and for turbine lubrication is being followed closely but at the present time there is no non-inflammable lubricant that is considered entirely suitable for the requirements. A fluid suitable for the regulating system may possibly be developed but using this fluid for regulation and oil for lubrication involves duplication of pumps and other equipment which would still further complicate the operation and require considerably greater attention in order to get the same reliable operation now obtained. In view of the above, the system of regulation and lubrication is being designed so that the steam turbine will be fully protected from fires, using the same method of control and lubrication as has proved entirely satisfactory in the past."

The Westinghouse Company describes a dual fluid system that has been developed (see sketch) in which a non-inflammable fluid (Aroclor) is used for all valve operations while oil is retained for actual governing and bearing lubrication. Aroclor is a stable compound of chlorine and diphenyl. Tests and operating experience prove that it is entirely satisfactory for use in hydraulic governor control and valve operating mechanisms. Aroclor has been applied to the governing systems on two new units now in operation and on one older unit. These machines are listed below:

Owner	Station	Rating
Philadelphia Electric Co.	Delaware, Phila.	30,000 kw
Public Service Elec. & Gas Co.	Burlington, N. J.	18,000 kw
Hawaiian Electric Co.	Honolulu, T. H.	10,000 kw

"This fluid will also be used as the governing medium on the new 165,000-kw unit for the Philadelphia Electric Company. Its application to the older 30,000-kw unit listed above necessitated some changes to separate the governing and lubricating systems. Many older units can be changed in a similar manner and information necessary to carry out such changes with a minimum of governor adjustment and trouble can be supplied.

"Aroclor has lubricating qualities comparable, to some extent, to those of oil. Laboratory tests indicate that the coefficients of friction under the same conditions (especially at the same viscosities) are practically equal. However, due to the fact that at higher temperatures the Aroclor viscosity drops more rapidly than that of oil, the Aroclor film in a bearing is probably not as stable as an oil film.

"No attempt has yet been made to use Aroclor as a bearing lubricant and at the present time no plans or studies are under way leading up to such an application. One instance is known of where Aroclor has been used as a lubricating and governing medium on a small central-station turbine, but results of the test are not available. The value of the reduced fire hazard is, without doubt, sufficiently important to justify a prolonged experiment of this kind. Furthermore, it seems evident that such experiments should be carried out by the interested owners of the turbines because it is only by continued exposure to routine operating conditions that real conclusions can be reached.

"Other fluids: In search for a less expensive fluid to serve as a governing and valve operating medium, a number of other fluids, mixtures and solutions have been investigated. The most promising of these are mixtures of the soluble oils and water. However, nothing has yet been found which fulfills the requirements as satisfactorily as Aroclor."

The General Electric Company says:

"For the past year thorough tests have been made on the lubricating properties and flash points of various fluids in an attempt to find a satisfactory substitute for oil. So far nothing has been discovered that would warrant prejudicing the safety of important turbine installations by a trial substitution for approved lubricating oil.

"On the other hand, designs have been perfected which so isolate the oil tanks and confine the high-pressure piping as to greatly reduce the chance of oil fires in modern installations. While the General Electric Company stands ready to furnish a separate control system actuated by a non-inflammable fluid separate from the lubrication system, the complexity of this arrangement is not recommended in preference to the arrangements adopted on the most recently built turbines."

In the system recommended by this company, all pumps, coolers and the tank are confined in a fire-proof compartment in the basement, below the turbine operating floor. The high-pressure piping is entirely enclosed within the front pedestal for the control mechanism and No. 1 bearing, and within the large size return piping for all other bearings. It is welded throughout its entire length.

An alternative design employs two complete tank and pump systems, one for the non-inflammable high-pressure control fluid and the other for the lubricating oil. No installations of this type have yet been made.

Joseph H. Keenan, since 1928 assistant professor of mechanical engineering at Stevens Institute of Technology, and well known as the author of the Keenan Steam Tables, has been appointed professor of mechanical engineering at Massachusetts Institute of Technology, Cambridge, Mass.

R. B. Mildon has been elected vice president of the Westinghouse Electric & Manufacturing Company in charge of engineering, manufacturing and service at the South Philadelphia works of that company.

Charles S. Gladden, formerly Director of Power and Maintenance, General Motors Corporation, Detroit, is now with the power department of the E. I. du Pont de Nemours & Co., Wilmington, Del.

John Hunter, consulting engineer, St. Louis, Mo., has recently been appointed Advisory Engineer for the Riley Stoker Corporation, Worcester, Mass.

Henry Kreisinger, Combustion Engineering Company, Inc., has been appointed chairman of a Smoke Prevention Committee formed by the American Boiler Manufacturers Association, and affiliated industries. This committee will co-operate with the Smoke Prevention Association and other bodies active in the work of smoke elimination.

STEAM ENGINEERING ABROAD

As reported in the foreign technical press

Fuel Research Station

Analogous to our U. S. Bureau of Mines Experiment Station is the Fuel Research Station of the British Government at Greenwich, Eng. This station has been in existence for some years and is equipped to carry on large scale experiments with laboratory precision. It works in close contact with coal field activities through a standing Survey Committee of coal operators and others. Among the problems now under investigation are suspension of coal in oil, gas making in horizontal retorts, smokeless fuels, oils from tar, cleaning of coal and pulverized coal burners. The station and its activities are described and illustrated at length in the July issue of *The Fuel Economist*, London.

New Fulham Power Station

This latest British power station, which should be completed late this year is laid out for six 260,000 lb per hr boilers, to operate at 625 lb pressure and 850 F steam temperature. Firing will be with multiple-retort underfeed stokers. Both economizers and air preheaters will be employed, the latter delivering air to the stokers at 400 F. The ashes will be handled hydraulically and boiler makeup will be supplied by evaporators. The initial generating capacity will consist of two 60,000-kw turbine-generators. A preliminary description appears in the July issue of *The Steam Engineer*, London.

Mechanical Stokers for Ships

Three sea-going train ferries are being built for the Southern Railway Company to ply between Dover and Dunkirk, as was commented on editorially in the August issue of COMBUSTION. This installation is subsequently described and illustrated in some detail in the August issue of *The Steam Engineer* of London. The boats will be equipped with multiple-retort stokers of the Taylor type under 250 lb Yarrow boilers of the straight-tube type. Preheated air of 350 to 400 F will be employed. The coal bunkers are arranged so that the coal feeds by gravity to the stoker hoppers. Ashes will be raked out at the back and disposed of by steam ejectors. The coal used approximates 21 per cent volatile, 9 per cent ash and 2 per cent moisture.

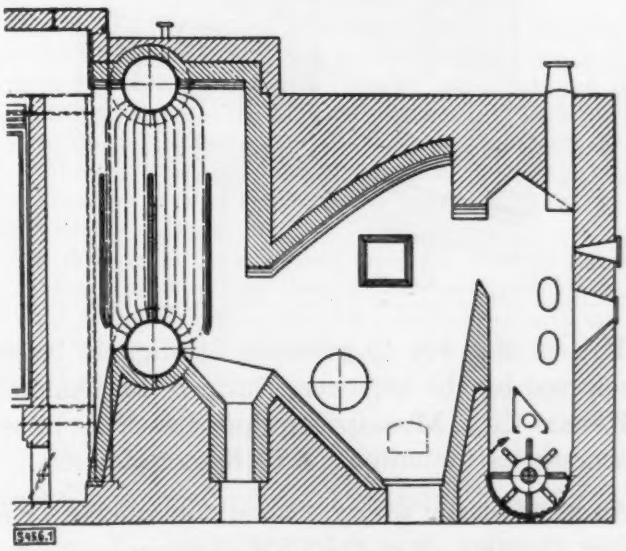
Another marine installation of stokers is described in the July 27 issue of *Engineering*. This is on the S.S. Shuntien, a cargo and passenger vessel constructed for the China Navigation Company. The ship contains two 7210 sq ft B & W marine-type boilers with tubular type air heaters and closed stokehold operating under forced draft. Each boiler is fired by an Erith-Roe six-retort underfeed stoker. From the operating floor to the top of stoker hopper is about $5^1/_2$ ft and the floor at the rear of the boiler is dropped slightly to provide for the removal of ashes which are raked out and handled by an ash ejector.

Blending Coals for Boiler Purposes

The Power Engineer (London) for August describes an unique blending plant that has just been put in operation at Wandsworth, Eng. There are three large storage hoppers, placed side by side, and each filled with a different coal by a drag line conveyor. A belt conveyor runs under the three hoppers and extends to the coal mixer which is of the worm type.

Mill Firing

An unique arrangement in which mill impellers are placed within the boiler setting, in advance of the furnace, is described in a recent issue of *Archiv Fuer Waermewirtschaft*. This is shown in the cross-section reproduced herewith. The top of the mill chamber is in direct communication with the furnace and serves as a separator for the mill, thus eliminating the usual separator, mill fan, duct work and burners. Drying is accomplished by heated air admitted to the shaft above



The mill is within the boiler setting

the mill. Sufficient fines are delivered through the opening from the mill chamber and ignite upon entrance into the furnace. The system is simple, cheap to install and is said to be particularly adapted to wet brown coals. On test, brown coal containing 50 to 53.5 per cent moisture, was burned at combustion rates of 7900 to 37,700

Btu per cu ft of furnace, the air temperature ranged from 410 to 752 F and high CO_2 was maintained, particularly at $^3/_4$ load. The boiler efficiency reported was 81 per cent at $^1/_5$ load, 88 per cent at $^3/_4$ load and 85.5 per cent at full load. At the most economical load the mill required 3.5 kwhr per ton of coal handled. This increased up to 12 kw per ton at $^1/_5$ load.

Loeffler Boiler Tests

The June 29 issue of *Engineering*, London, contains the results of tests by Professor Josse on one of the 132,300 lb boilers at the Caroline Station in Czechoslovakia which operate at 1850 lb pressure and 932 F. Pulverized coal is burned. An overall efficiency of 87 per cent was attained. The turbine driving the steam circulating pump consumed 3 per cent of the total output but much of this was recovered in heat returned to the feedwater. A constant steam temperature was maintained over rapid load swings of three minutes from 72,500 lb to 132,250 lb of steam per hour. The make-up averages 7 to 8 per cent and is taken from the river without any treatment. After 2000 hr operation some of the superheater tubes were cut open and found to be free from deposit. Reference is also made to the new station of the Moravian-Silesian Electricity Company which contains three 495,000 lb per hour Loeffler boilers.

Brimsdown Station

There has lately been completed an extension to the well-known Brimsdown Station of the North Metropolitan Electric Company (England), which comprises four 200,000 lb Wood steam generators fired with pulverized coal. The Raymond Mills are designed to handle coal of 17 per cent ash and 18 per cent moisture. Steam conditions are 314 lb and 784 F at the superheater outlet and preheated air at 500 F is supplied by a regenerative type air heater. An elaborate system of gas washing is installed. On a test lasting 168 hr an overall efficiency of 87.58 per cent was obtained. Two new turbine units have been installed, one of which is a 25,000-kw, 3000 rpm Parsons machine supplying current at 33,000 volts. An extension and detailed description of this extension to Brimsdown will be found in the June 8, 15, 22 and 29 issues of *Engineering*, London.

New Patent Laws in Germany

Political changes in Germany have resulted in a number of changes in her patent laws and practice, according to *Engineering* of August 10, 1934. These concern principally patent attorneys or agents, procedure in civil suits and infringement. All representations before the Patent Office must be made by registered patent attorneys and Jews are excluded from becoming patent agents (with certain exceptions) in the future. Under the new procedure it is now possible to examine the parties before the court and no party will be permitted to lead the court astray through mis-statements or to abuse its procedure by protraction of the action. Contrary to English practice, there is no rule in Germany that every claim is a disclaimer, but a German patent may cover much more

than the literal contents of the claim. In a recent decision it was held that the patentee is liable for damages if he has warned a possible infringer in good faith and it is afterward decided that the patent was not infringed. Therefore, great care should be taken in warning possible infringers in Germany.

Economizer Explosions

The results of an investigation into the causes of cast iron economizer explosions as conducted jointly by The Association of Large Boiler Owners and the Association for the Supervision of Power Economics in the Ruhr mines is reported in a recent issue of *Zeitschrift des Vereines deutscher Ingenieure.* Economizer failures were traced to two causes, namely gas explosions and sudden non-uniform stress due to variations in load. An explosive mixture is not formed until there is one part air to two of gas (consisting of approximately 30 per cent CO_2, CO and H_2 and 70 per cent N_2). The air may reach this proportion through leakage or a sudden change in draft. Tests showed that the temperatures and flow through the various tubes are far from uniform and that sudden changes in load may set up stresses sufficient to cause failure.

A Compact Arrangement of Air Heaters

Fig. 1 shows a proposed arrangement of four air heaters to control the temperature at the cold end. Numbers 1, 2, 3 and 4, are arranged in groups of two so that 1 and 2 are together and 3 and 4 are together, with a space between the two groups. This space between groups is used, between heaters 1 and 3 at the top, for the entrance of the cold air; the space between 2 and 4 is used for the exit of the hot air. Heaters 1 and 2 are connected in series, as are also 3 and 4, whereas the groups of air heaters 1–2 and 3–4 are connected in parallel.

Fig. 2 is a diagrammatic representation of the flow through the air heaters. Air enters from the left top, divides, traveling downward through heaters 1 and 3 and at the bottom of these heaters outward through connecting ducts and inward into the bottom of heaters 2 and 4. Traveling upward through 2 and 4, it then leaves these heaters at the space between them and discharges downwardly to the burners.

Gas flows from the right and divides to heaters 2 and 4, flowing down through these heaters out through the bottom into the hopper shaped connecting ducts and thence upwardly through heaters 1 and 3, discharging at the top to the stack.

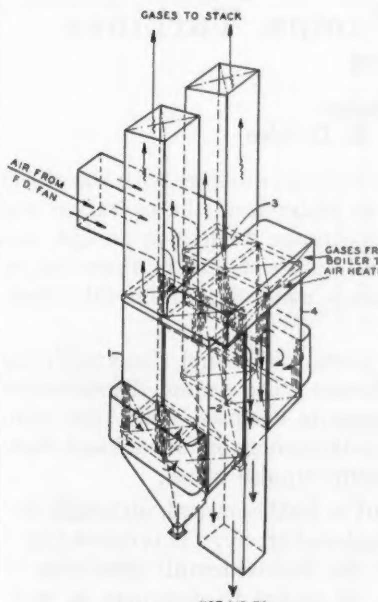


Fig. 1

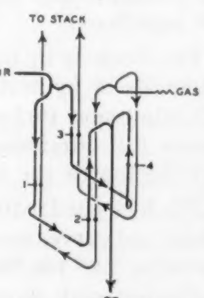


Fig. 2

Arrangement of air heaters to control the temperature at the cold end

New Code for Unfired Pressure Vessels

A code for the design, construction, inspection and repair of unfired pressure vessels for petroleum liquids and gases has been completed by joint committees of the American Society of Mechanical Engineers and the American Petroleum Institute. The joint committee was at work for more than two and one-half years and covers vessels for flammable liquids or gases not covered by the A.S.M.E. Rules for unfired pressure vessels. The code is now available in pamphlet form.

REVIEW OF NEW BOOKS

Any of the books reviewed on this page may be secured from Combustion Publishing Company, Inc., 200 Madison Ave., New York

Principles of Engineering Thermodynamics

By Newton C. Ebaugh

This book is written for those students of engineering, in technical schools or in the practice of their profession, who wish to acquaint themselves with the essential principles of the science of thermodynamics as it applies to much of our engineering processes and equipment.

The material covered in this book, as indicated by the chapter headings, is as follows: Thermodynamics and Energy; State of a Fluid and Changes of State; Energy Equations; Gas Properties and Processes; Cycles and Available Energy; Entropy; Vapor Properties and Processes; Gas Mixtures—Gas and Vapor Mixtures; Flow of Fluids; Vapor Power Cycles; Internal Combustion Cycles; Gas Compression; Refrigeration Cycles.

It has been the author's endeavor to so organize the subject matter of the book that the sequence of thought as revealed in the first eight chapters will convey to the student the principal steps in the development of the science in the most logical and concise manner. These chapters will lay the foundation for many different applications in the field of engineering. A few of these applications are discussed in the latter chapters.

The book contains 190 pages, size $8^3/_5 \times 10^7/_8$. Price $2.25.

The New Background of Science

By Sir James Jeans

The subject matter of this book reminds us of that old eastern fable of the blind philosophers who, each from his viewpoint, undertook to describe an elephant. You will remember that one, having hold of the elephant's tail, declared the beast to resemble a snake; another, feeling of the behemoth's leg, insisted that he was like a tree trunk; and so on.

So with science. As Jeans points out, our assumptions, both of fact and theory, regarding natural phenomena are directed by the viewpoint from which we make the observations on which these assumptions are based. Science, says Jeans, was until recently thought of as a "common sense" view of nature. The scientist believed that there was no great difference between appearance and reality. The rules that governed the world of mechanics, chemistry, electricity and astronomy, as determined on such a basis, were positive; as absolute, for example, as the oft-quoted second law of thermodynamics.

Yet, further observation into the ultimate realities discloses such an apparent lack of determinism in the characteristics and behavior of the building bricks of the universe—such as electrons and photons—as seemingly to baffle investigation and theory. Relativity, of course, is the essence of these concepts; its theory enables reconciliation of many otherwise conflicting observations and conclusions. The later quantum theory accounts for many other observed phenomena at variance with the concept of all-pervading ether. Still other manifestations remain perplexingly variable. Are electrons bullets of electrified matter, quanta? Or are they waves? Both characteristics are presented. But if waves, then waves of what? And how account for characteristics incompatible with anything wave-like?

Difficulties such as these arise because of the limitations of our senses and because of the fact that the world of science cannot be explored except "by tramping over it and disturbing it; and our vision of nature includes the clouds of dust we ourselves kick up. We may make clouds of different kinds, but the uncertainty principle shows that there is no way of crossing the desert without raising a cloud of some kind or other to obstruct the view."

It is by imparting to the reader an understanding of this viewpoint that Jeans affords the reader of this book a clearer picture of the present knowledge and position of science. And, whatever dust clouds may be stirred up by the investigations of the scientist, there are none created by Jeans in his exposition of the subject. One of the marvels of the book is the extraordinary clarity with which the author makes understandable to the average intelligence the complexities necessarily encountered. To all who care to be informed as to the newest theories and methods of attack on the frontiers of pure science this book of Sir James Jeans will be found to be fascinating and memorable.

The New Background of Science contains 301 pages, including index, size $5^1/_2, \times 8^1/_4$, and is bound in dark blue cloth. Price $2.50.

Centrifugal Pumps, Turbines and Propellers

By Wilhelm Spannhake
Translated by John B. Drisko

A text covering fundamental principles with the object of enabling the reader to understand the operation and characteristics of such hydraulic devices as pumps, turbines and propellers, and to assist him in their design. Only centrifugal machines using incompressible fluids are considered.

The book is in four parts. The first deals with the essentials of hydromechanics; the second discusses the full admission turborunner in enclosed flow; the third covers full admission turborunners in unenclosed flow; and the fourth the modern impulse wheel.

Much of the treatment is mathematical, although diagrams and curves are employed freely. It is essentially a textbook for teaching the fundamental principles of hydraulics and should be useful to designers as well. The translation grew out of Professor Spannhake's lectures at the Massachusetts Institute of Technology.

The book contains 328 pages, 6×9 in. and 182 illustrations. Price $5.00.

EQUIPMENT SALES

Boiler, Stoker, Pulverized Fuel

As reported by equipment manufacturers of the Department of Commerce, Bureau of the Census

Boiler Sales

Orders for 88 water-tube and h.r.t. boilers were placed in July

	Number	Square Feet
July, 1934	88	216,640
July, 1933	144	455,822
January to July (inclusive, 1934)	524	1,461,696
Same period, 1933	527	1,602,072

NEW ORDERS, BY KIND, PLACED IN JULY, 1933–1934

	July, 1933		July, 1934	
Kind	Number	Square Feet	Number	Square Feet
Stationary:				
Water tube	90	380,090	35	145,802
Horizontal return tubular	54	75,732	53	70,838
	144	455,822	88	216,640

Mechanical Stoker Sales

Orders for 187 stokers, Class, 4* totaling 42,658 hp were placed in July by 60 manufacturers

	Installed under			
	Fire-tube Boilers		Water-tube Boilers	
	No.	Horsepower	No.	Horsepower
July, 1934	119	16,724	68	25,934
July, 1933	110	15,047	67	27,062
January to July (inclusive, 1934)	605	80,972	285	113,971
Same period, 1933	501	66,226	248	89,640

* Capacity over 300 lb of coal per hr.

Pulverized Fuel Equipment Sales

Orders for 15 pulverizers with a total capacity of 59,500 lb per hr were placed in July

STORAGE SYSTEM

	Pulverizers				Water-tube Boilers		
	Total number	No. for new boilers, furnaces and kilns	No. for existing boilers	Total capacity lb coal per hour for contract	Number	Total sq ft steam-generating surface	Total lb steam per hour equivalent
July, 1934	..	..	..	......	..	....	....
July, 1933	..	..	..	......	..	....	....
January to July (inclusive, 1934)	2	1	1	46,000	..	....	....
Same period, 1933	2	..	2	60,000	2	37,000	325,000

DIRECT FIRED OR UNIT SYSTEM

	Pulverizers				Water-tube Boilers		
July, 1934	15	12	3	59,500	12	61,100	501,200
July, 1933	9	9	..	92,600	8	69,733	829,600
January to July (inclusive, 1934)	46	34	12	312,110	35	269,575	2,645,900
Same period, 1933	43	32	11	252,840	35	244,406	2,312,960
					Fire-tube Boilers		
July, 1934	..	..	..	....	..	....	......
July, 1933	..	..	..	....	..	....	......
January to July (inclusive, 1934)	4	..	4	4,800	5	7,500	41,000
Same period, 1933	9	2	7	7,700	10	14,150	64,900

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